

THE EFFECTS OF LUBRICANT CONTAMINATION ON ROLLING BEARING PERFORMANCE

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Abstract

Recent improvements in the quality and manufacture of steels have meant that components in rolling contact are largely free from materials defects. It is therefore damage generated during the operating life of the component that is the main source of failure. This damage is frequently caused by the overrolling of debris particles in the lubricant. This damage can lead to premature failure of the component by abrasive wear or surface initiated fatigue.

The thesis begins with an investigation into the mechanisms of debris particle entrainment and overrolling in a lubricated contact. Using the principles of optical elastohydrodynamics, where one contacting element is replaced by glass, particles have been studied entering and fracturing or deforming in a contact.

The surface damage caused by these particulates has been closely analysed and related to the properties of the debris material. The hardness of ductile materials and the fracture toughness of brittle materials were found to be important parameters in controlling the level of surface damage.

Plastic contact finite element analysis has been used to analyse the stresses and deformations associated with the indentation process. Residual stress information from various model indentations has been coupled with a rolling contact fatigue model to predict the lives of the damaged surfaces.

When surface dents are below a fatigue initiating threshold, failure is by excessive material loss. This is a process of three body abrasive wear, and has been closely studied. Some fundamental experiments have been performed using a simple ball on disk contact operating under varying degrees of sliding. Abrasive particles have been shown, depending on the conditions, to tumble or imbed in one surface and scratch the other.

→ The information from these tests has been used to develop a model of the abrasive wear of rolling element bearings by contaminated lubricants. The model is based on predictions of the number of particles entering a contact, their sliding distance, the cross sectional area of material they displace, and the fraction of that area removed as wear. Rolling bearings have been tested under a variety of contamination conditions. Measurements of abrasive mass loss have been compared to predictions from the model.

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Nomenclature

A	cutting area of an abrading particle
A_1	cross sectional area of scratch bounded by shoulders and mean surface line
A_2	cross sectional area of scratch shoulders
A_3	cross sectional area of scratch below mean surface line
a	radius of contact circle (circular point contact)
a	semi-minor axis of contact ellipse
a_{α}	critical crack length
b	semi-major axis of contact ellipse
D	ball diameter
d	sliding distance
d_{av}	average sliding distance across a bearing contact
d_m	bearing pitch diameter, $1/2(d_o+d_i)$
d_i	inner raceway diameter
d_o	outer raceway diameter
E	Young's modulus (modulus of elasticity)
E^*	reduced modulus, $1/E^*=(1-\nu_1^2)/E_1+(1-\nu_2^2)/E_2$
e	eccentricity of loading (of a rolling bearing)
f	fraction of a displaced area removed as a wear chip
G	non-dimensional materials parameter, $2\alpha E^*$
G	bulk modulus
g	fraction of the sliding distance accommodated at the surface under consideration
H	non-dimensional film thickness parameter, h/R_x
H	hardness
h	lubricant film thickness
h_c	central lubricant film thickness
$h_{\min}$	minimum lubricant film thickness
K_{IC}	fracture toughness
k	limiting shear stress
k	ellipticity parameter, b/a
k	wear coefficient
L	life
M	wear mass
n	refractive index
n	number of particles entering a single contact
P	load
P_d	bearing diametral clearance, d_o-d_i-2D
P_y	load at first yield
p	pressure

p_0	maximum contact pressure
p_H	hydrostatic pressure
p_m	mean contact pressure
$Q_{\max}$	maximum ball load (at $\psi=0$)
Q_ψ	ball load at azimuth angle ψ
R_a	centre line average (CLA) roughness
R_x	effective radius in the x -direction
R_y	effective radius in the y -direction
r_f	radius remote from contact (fringe position)
r_i	inner raceway groove radius
r_o	outer raceway groove radius
S	number of cycles
T	force in direction tangential to motion (traction force)
T_p	force on a particle in direction tangential to motion
U	non-dimensional speed parameter, $P/2E^*R_x$
u	surface velocity in the direction of motion, x
V_{swept}	volume of lubricant swept out
W	non-dimensional load parameter, $\eta_0 u / 2E^*R_x$
x	concentration of debris, g/l
x	Cartesian co-ordinate in direction of local axis x
Y	yield stress
Y	elastic depth recovery of an unloaded indentation
y	cartesian co-ordinate in direction of local axis y
z	cartesian co-ordinate in direction of local axis z
z	number of balls in a ball bearing
Γ	conformity factor, $b^2 E^* / 4\mu p_0 R^2$
α	pressure viscosity coefficient
α	half-width of an indentation
α	abrasive grit orientation angle
δ	deflection of a point
δ	depth of an indentation
δ	size of abrasive particle
δ_y	deflection at first yield
ϵ_x	strain along co-ordinate direction x
ϵ_y	strain along co-ordinate direction y
ϵ_z	strain along co-ordinate direction z
ϕ	semi prow angle of an abrasive particle or cutting tool
ϕ	osculation ($=D/2r$)
γ	location of pure rolling lines in a contact (at $y=\pm\gamma b$)
η	viscosity
η_0	viscosity at contact inlet conditions
λ	ratio of film thickness to composite RMS roughness (lambda ratio)
μ	friction (or traction) coefficient
θ	attack angle of an abrasive particle or cutting tool
ρ	density
$\Sigma\rho$	curvature sum

$F(\rho)$	curvature difference
σ	root mean square (RMS) roughness
σ_f	fracture strength
σ_x	stress along co-ordinate direction x
σ_y	stress along co-ordinate direction y
σ_z	stress along co-ordinate direction z
σ_z	axial stress
σ_r	radial stress
σ_θ	hoop stress
τ_{xy}	shear stress in plane xy
τ_θ	shear stress at angle θ
τ_{max}	maximum absolute shear stress
τ_u	fatigue shear stress limit
σ_1	principal stress 1
σ_2	principal stress 2
σ_3	principal stress 3
τ_1	principal shear stress
ν	Poisson's ratio
ω	angular velocity
ξ	creep or slip ratio
ξ_0	creep or slip ratio at the axis of the ball, $y=0$
ψ	ball position in a bearing (azimuth angle)

Subscripts and Superscripts

0	maximum (for pressure)
1	element or surface 1
2	element or surface 2
av	mean value
b	ball
i	inner race
o	outer race
m	mean value
max	maximum value
x	direction of local axis x
y	direction of local axis y
z	direction of local axis z
ψ	occurring at ball azimuth angle ψ
*	for a strip (for microslip)
*	reduced value (for moduli)

CHAPTER 1

INTRODUCTION

The expected life of rolling element bearings has increased over recent years. Improvements in the quality and manufacture of steels have meant that components in rolling contact are largely free from materials defects. The most common cause of failure now tends to be from damage initiated at the bearing surface. This damage is usually the result of the overrolling of debris particles in the lubricant. The surface damage can act as a stress raiser and lead to fatigue; or, over a period of time, to excessive material loss by abrasive wear. The bearing industry has witnessed a shift from sub-surface initiated failure, at material inclusions or voids, to surface initiated failure, at debris damage sites.

The mechanisms by which overrolled particles cause damage to, and lead to material loss from, bearing surfaces are not always well understood. This study attempts to relate surface damage to debris material and size; and then use this information in the determination of fatigue life and in understanding rolling bearing abrasive wear.

This chapter is presented as an introduction to the problem. Reviews of existing work are presented with the chapters as appropriate.

1.1. The Rolling Element Bearing

Friction was a barrier to the technological development of early man, how could large objects be moved by muscle power alone? Man, with the invention of the wheel, found it easier to move these objects when they were supported on rollers. Overcoming friction (see Figure 1.1) allowed the transport of food, building materials, and later people; it contributed to the development of civilisation itself.

These early developments in the reduction of friction by rolling and later by lubrication were the first steps towards the development of the rolling element bearing. Figure 1.2 shows a modern rolling bearing. Load is transmitted between the two bodies in relative motion, the inner and outer raceways, by the rolling elements interposed between them.



Figure 1.1. Early man overcoming friction.

The availability of rolling element bearings has had a tremendous impact on the engineering design of this century. Ball and roller bearings are used in machine elements throughout the world; from bicycles and shopping trolleys, to aircraft gas turbines and steel rolling mills.

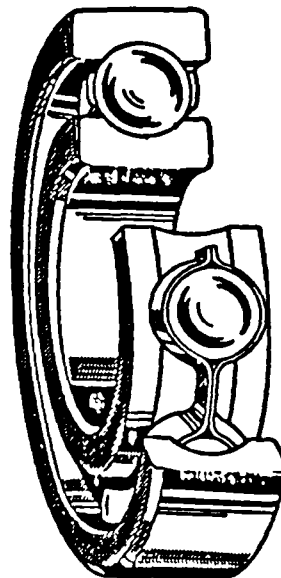


Figure 1.2. A modern rolling element bearing. Load is transmitted and friction reduced by interposing rolling elements between the surfaces in relative motion.

1.2. The Development of the Rolling Bearing - Improvements in Life

The first real improvements in the life of rolling bearings came in the late 1920's. These developments were based on the pioneering studies of Stribeck, Goodman, and Palmgren. Stribeck (1907) carried out experiments to determine the load capacity of bearings at various speeds. He tested the mechanical properties of bearing steels. He derived an expression relating ball load to applied load and verified many of Hertz's expressions for elastic contact. Goodman (1912) empirically related load carrying capacity to speed and bearing life. Lundberg and Palmgren's formula for bearing fatigue life, published in 1923, became a standard for bearing manufacturers.

At around the same time a textile mill employee, Sven Wingquist, noted the high proportion of bearing failures associated with mounting. He designed bearings without filling slots and the self-aligning bearing. His designs showed rapid market success and marked the birth of a new company, A.B. Svenska Kullagerfabriken or SKF.

A real breakthrough in the understanding of rolling bearing lubrication came with the analytical studies of Grubin (1949), who incorporated the pressure-viscosity characteristics of lubricants into the fluid flow equations of Reynolds assuming Hertzian elastic deformation. He predicted realistic film thicknesses which were comparable with the roughness of the bearing surfaces. Later Hamrock and Dowson (1976) developed a more rigorous solution by allowing elastic deformation to vary also.

It was developments in the quality of the bearing steels which heralded the next improvement in bearing endurance. Manufacturing process, such as continuous casting, uphill teeming, and vacuum degassing improved the steel quality and reduced the number of inclusions and the oxygen content.

Figure 1.3 below demonstrates how these developments have led to a dramatic increase in the nominal lives of rolling bearings.

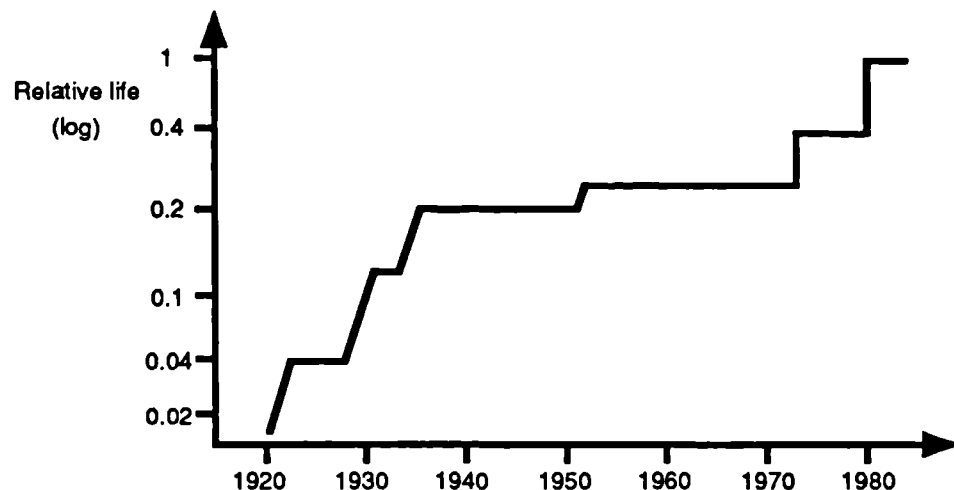


Figure 1.3. Diagram demonstrating the increase in nominal bearing life over this century. Lives are plotted as a fraction of a current rating; note the logarithmic scale.

These developments in rolling bearing design and manufacture have removed many common sources of failure. We are now left with failure from damage initiated during the bearing operating life. The major cause of such damage is the overrolling of particles in a contaminated lubricant.

"The presence of dirt or grit in ball bearings is responsible for over 90 percent of all bearing failures; where bearings are kept clean during mounting, are lubricated with clean oil and are protected by recommended closures, no difficulty should be experienced from this source."

- General Motors Corporation (1971)

Solving this problem, either through lubricant treatment or clever bearing design, will result in longer bearing lives.

1.3. Source and Nature of Lubricant Contamination

A contaminant is any foreign or unwanted substance that can have deleterious effects on system operation, service life, or reliability. Material contaminant can exist as a gas, liquid, or solid. Gases can be in the dissolved state or in the form of bubbles and slugs; liquids can be free, dissolved, or emulsified; and solids are in the form of particles.

It is the solid contaminating particles that are of interest in this study. There are three sources of contamination in any operating machinery;

- (i) **Implanted Contamination.** Particles may pre-exist in the system being introduced during manufacture, handling, packaging, dispatching, or assembly.
- (ii) **Generated Contamination.** Particles may be produced during the operation of the system or of secondary systems. By far the most common form of generated particulate debris is caused by wear of gears, bearings, or other components in rolling or sliding contact.
- (iii) **Ingested Contamination.** Particles may enter the system from the operating environment through seals, breathers, filler caps, or during maintenance operations.

To define the extent of a contamination problem it is necessary to analyse the lubricant from the operating machinery. A full analysis must incorporate measurements of particle concentration, size distribution, and material. The first two measurements are relatively easy, since measurements can be made on a large scale using optical methods (light scattering or obscuration). The determination of debris materials is more complex; some methods used include, spectroscopy, IR emission, and ferrography. In this study scanning electron microscopy and EDAX were used. As with all batch sampling techniques, the results are only meaningful if the sample accurately reflects the lubricant which will become entrained into contacting components.

In some preliminary studies for this work, oil samples were filtered through 0.8 μ m membrane filters. The membranes were then washed and dried and mounted under a microscope. Estimates of the debris concentration were made by particle counting. The membrane was then gold coated and viewed on an SEM. The SEM and EDAX were used to pick out and magnify individual particles and determine the elements present. Table 1.1 shows the results of these analyses and compares them to the data of other investigators.

Investigator	Sample source	Concentration (estimated)	Size Range	Materials present
This author	paper mill	0.4g/l-	0-150 μ m	Cu, Sn, silicates
This author	motor vehicle sump	1-2g/l	0-250 μ m	C, Fe
SKF ERC (Berghini 1992)	various bearing lubricants	0.3-1.5g/l	0-250 μ m	Fe, Al, Cu, Sn, SiC, sand
Admiralty Engineering Lab	various	-	0-100 μ m	Al, Ag, Cr, Zn, Fe
Eleftherakis & Khalil (1990)	motor vehicles	0.3g/l	0-120 μ m	Al, Cu, Fe, Pb
Loewenthal & Moyer (1979)	aircraft gas turbines	1-2g/l	0-200 μ m	Fe, C, silicates

ref.

Table 1.1. Contaminating particles found in lubricant samples from various sources.

Steel particles are frequently present, possibly from worn gears or other contacting components. Aluminium, bronze and copper may have originated from worn bushes or housings. Silicates are likely to be in the form of dust or sand from casting moulds or ingested into the system from the environment. Hard ceramic materials, like silicon carbide, may be left over from grinding operations

1.4. Relating Lubricant Contamination to Bearing Performance

Figure 1.4 schematically shows the relationship between lubricant contamination and rolling bearing performance.

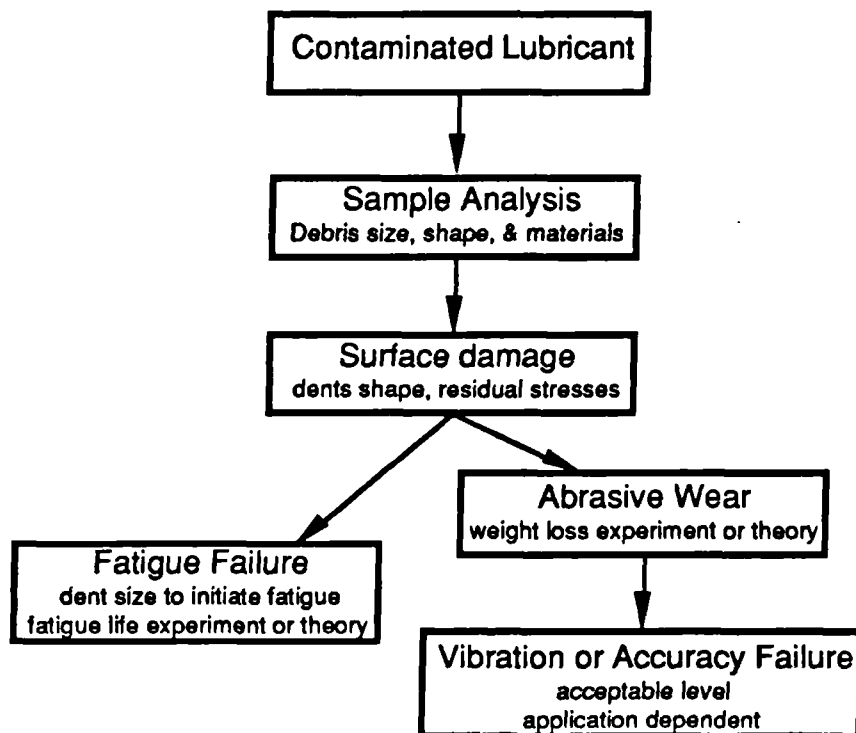


Figure 1.4. Schematic diagram of the relationship between lubricant contamination and rolling bearing performance.

It is instructive to consider the information that would be required to fully understand the problem. A comprehensive study would require addressing each of these steps;

- (a) The contaminants present in a lubricants. Some experience of the type of contaminating particles present for a given application is required. The information would need to include debris material, size, and concentration. So, for a given lubricant one would need to perform a full analysis, or predict the contamination level from the results of an analysis of a sample from a similar application.
- (b) The entry of particles into a contact. If a contaminant is present in the lubricant, will it enter the component contact? One aspect of this is the presence of filters or screens. Another aspect is whether particles presented at the contact inlet will become entrained.
- (c) The effects of overrolling contaminants. Given that a particle enters a contact, it is necessary to determine whether surface damage will result. This will depend on the material properties of the debris and surfaces, and on the size of the particles.

- (d) Surface damage and fatigue. Given a damaged surface (or a quantified concentration of debris) the next step is to determine if rolling contact fatigue will result. This may be approached theoretically using a rolling contact fatigue model which accounts for local stress effects. Or experimentally by testing bearings in contaminated lubricant or with pre-indented elements.
- (e) Surface damage and wear. If the surface damage is not sufficient to initiate fatigue, then the failure mechanism may be one of abrasive wear. Excessive wear leads to increased clearance in the bearing and so to excessive vibration. There is a requirement to gain an understanding of how debris causes wear, and to be able to specify some maximum bearing clearance for a given safe application.

1.5. Aims of the Project

Contamination and its effects on the performance of contacting components clearly requires an extensive study. Only a small part of the problem can be covered within the framework of a PhD project. It is sections (b), (c), (d), and (e) from above, which have been considered in this study. The aims have been;

- (i) To understand how debris behaves in and around an elastohydrodynamic contact.
- (ii) To develop predictions of the surface damage caused by the overrolling of debris particles.
- (iii) To relate surface damage to rolling contact fatigue life. This has been a theoretical study only.
- (iv) To understand the mechanism of abrasive wear in a concentrated contact.

The thesis covers these objectives in approximately this order. Chapter 2 describes the work on debris visualisation in an elastohydrodynamic contact using an optical contact rig. Chapter 3 outlines experiments performed to quantify surface damage from overrolled particles. Chapter 4 covers the results of finite element calculations to predict dent shapes and residual stresses. Chapter 5 uses this information in a theoretical study of the rolling contact fatigue of indented surfaces. Chapters 6, 7, and 8 detail the experimental and theoretical studies of abrasive wear in concentrated contacts.

1.6. Concluding Remarks

A major cause of bearing failure results from the overrolling of lubricant debris. The work described in this study is aimed at *improving the understanding of the relationship between lubricant particulate contamination and rolling bearing failure. The study has concentrated on three aspects of the problem; firstly, how debris enters lubricated contacts and causes surface damage; secondly, how this surface damage can effect fatigue life; and lastly, how debris can cause abrasion of the contacting elements.*

CHAPTER 2

DEBRIS IN AN ELASTOHYDRODYNAMIC CONTACT

2.1. Introduction

The main body of industrial research into contamination in lubricants has been aimed towards defining of the relationships between contamination level and fatigue life. This is, of course, supremely important for a bearing user, who knowing this can then seek to condition lubricants to be free of a dangerous concentration of debris particles. However, it is also instructive to understand how different types of contaminating particle alter the lubrication regime, and how this effects the performance of a lubricated contact.

This chapter has been devoted to a fundamental experimental study of the behaviour of debris particles in the region of an elastohydrodynamic contact. The entry and deformation of particles are discussed and this information is related to surface damage and contact failure. The chapter begins with a general discussion on elastohydrodynamic lubrication and a review of existing work on the behaviour of debris particles in this regime.

2.1.1. Lubrication Regimes

When surfaces are in contact the asperities touch and because the load is supported over a small area, high pressures and plastic flow result. The asperities may locally weld and form junctions. If relative motion occurs these junctions are broken and reform. High friction coefficients and surface damage results. Engineering components operating under these conditions would not last long. The purpose of lubrication is to separate these surfaces by interposing a layer of low shear stress material. In general, the thicker this lubricating film the better, since the surfaces will be well separated and friction kept low. There are five main regimes of lubrication;

- (a) Dry Friction. Oxide films and adsorbed layers cover surfaces in contact. These films offer some slight protection, but generally seize at moderate loads and speeds.
- (b) Hydrodynamic Lubrication. The two rubbing surfaces have geometrical wedge shaped separations; relative motion between the two causes lubricant to be dragged in between them.

- (c) **Hydrostatic Lubrication.** In this process the contacting surfaces are separated by lubricant film which is fed from an external high pressure source.
- (d) **Boundary Lubrication.** The contacting components are partly separated by a low shear strength chemical layer which is normally adsorbed or adhered onto the surfaces. The lubricant layer is thin and considerable solid contact occurs. Chemically active compounds (usually sulphur or phosphorous compounds) are added to the lubricant to react with the surfaces and produce these boundary films.
- (e) **Elastohydrodynamic Lubrication.** When the two opposed surfaces contact over a small area, the contact pressures are high and result in distortion of the bodies and an increase in the viscosity of the lubricant.

Table 2.1 summarises these regimes and gives some typical friction coefficients and applications.

Lubrication Regime	Typical Film Thick, μm	Film Formation	Friction Coefficient, μ	Examples
Dry Friction	0.001-0.01	None	>0.5	-
Hydrodynamic	1-100	Oil dragged in	0.007-0.03	journal bearings
Hydrostatic	1-100	Oil pumped in	0.007-0.03	machine tool spindles
Boundary	0.001-0.05	Reactive additives	0.08-0.3	metal cutting
Elastohydrodynamic	0.1-1.0	Oil dragged in η increase	0.001-0.1	ball bearings, gears

Table 2.1. Summary of the five main regimes of lubrication with typical film thicknesses and friction coefficients.

Many practical lubrication systems operate under a mixture of regimes. At start-up or shut down the speed drops. At low speeds the film decreases and more surface contact occurs. More of the load is supported by the asperities and shear occurs at asperity junctions, as a result friction increases, at higher speeds the film separates the surfaces and the shear shifts from the asperities to the fluid and the friction drops. As the film thickens further the friction increases slightly; this is caused by the increased volume of lubricant in the contact which is being sheared.

A useful parameter in the study of surface contact and lubrication is the specific film thickness, or lambda ratio, λ . This is the ratio of the minimum film thickness to the composite RMS roughness, σ_c of the two surfaces, and is a measure of the amount of surface contact.

$$\lambda = \frac{h_0}{\sigma_c} \quad (2.1)$$

where,

$$\sigma_c = \sqrt{\sigma_1^2 + \sigma_2^2} \quad (2.2)$$

Figure 2.1 shows how the friction coefficient varies with lambda ratio and lubrication regime; the annotation shows where the lubrication regimes fall. This lambda ratio is a critical operating

parameter and has been found to correlate with wear (as contacting asperities plough the opposing surface), fatigue (surface contact induces high local stresses which may initiate crack growth), and scuffing (surface contact leads to high local temperatures and film failure).

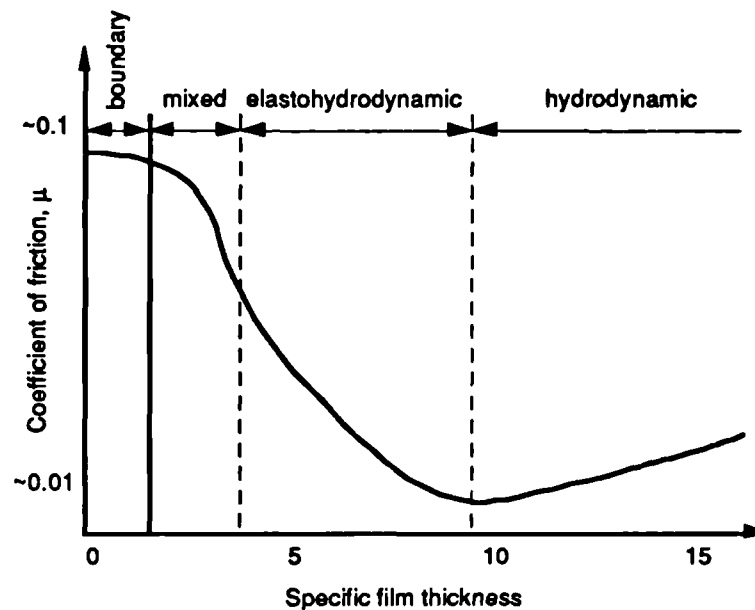


Figure 2.1. Schematic diagram of the variation of coefficient of friction, μ with specific film thickness, λ .

2.1.2. Elastohydrodynamic Lubrication

A clue to the meaning of the term, elastohydrodynamic lubrication (ehl) can be found in its name; *elasto* - distortion, *hydro* - fluid, and *dynamic* - moving. When two contacting surfaces are non-conforming, then the load is spread over a very small area (for example gear teeth, ball bearings, cam and tappets). The high contact pressures which results have two important effects; the contacting surfaces flatten and the viscosity of the lubricating oil increases considerably.

As the two bodies contact the lubricant is subjected to high pressures the viscosity increases exponentially (Blok 1952). This prevents the lubricant from escaping and results in a separating oil film between the bodies. A central flat region develops in the contact. As the fluid attempts to exit the contact, the pressure drops radically and hence the viscosity of the lubricant decreases; the lubricant flow rate therefore markedly increases. To maintain continuity at this increased flow rate the area must drop; hence the contact constricts. Figure 2.2 shows a contour map of an ehl point contact with the central plateau and horseshoe constrictions to the rear and sides.

Hamrock and Dowson (1976) developed a solution to the ehl problem. They numerically iterated the Reynolds equation with relations for elastic deformation, and for pressure-viscosity and pressure-density characteristics of the lubricant. The authors used the solution for a wide range of lubricants, materials, and operating parameters to determine empirical relationships for

the ehl film thickness. Five dimensionless parameters are defined; load, W , speed, U , ellipticity, k , materials, G , and film thickness H ¹;

$$W = \frac{P}{2 E^* R_x^2}, \quad U = \frac{\eta_0 u}{2 E^* R_x}, \quad k = \frac{b}{a}, \quad G = 2\alpha E^*, \quad H = \frac{h}{R_x} \quad (2.3)$$

where P is the load, E^* the reduced modulus, and R_x the combined radius in the rolling direction, η_0 is the viscosity at the contact inlet, u is the mean surface speed, α is the pressure viscosity coefficient, a and b are the semi-minor and semi-major axis of the contact ellipse, and h is the film thickness.

The film thickness parameter is then described in terms of the other four parameters. Two relationships were developed, one for the central film thickness, H_c and one for the minimum film thickness, H_{min} ;

$$H_{min} = 3.63 U^{0.68} G^{0.49} W^{-0.073} (1 - e^{-0.68 k}) \quad (2.4)$$

$$H_c = 2.69 U^{0.67} G^{0.53} W^{-0.067} (1 - 0.61 e^{-0.73 k}) \quad (2.5)$$

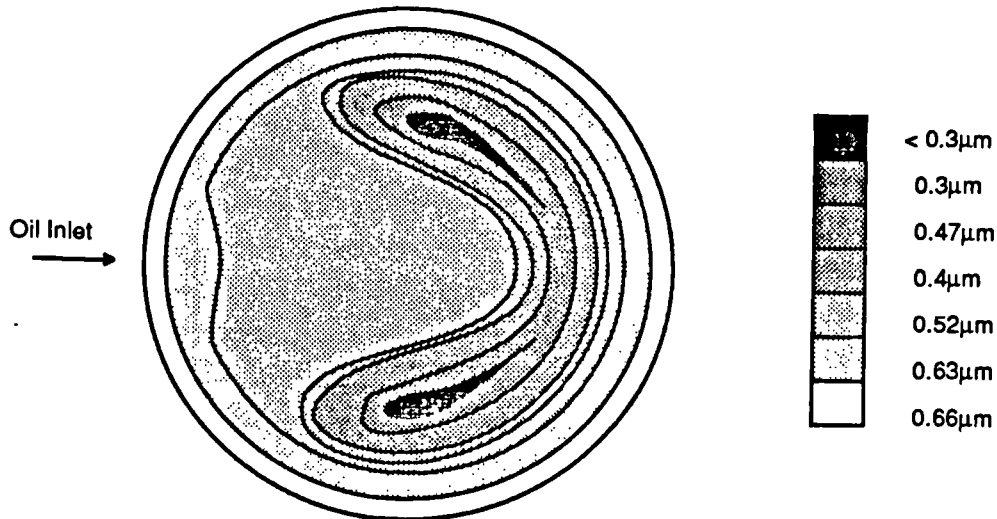


Figure 2.2. Diagrammatic contour map of a typical elastohydrodynamic film for a circular point contact.

Many experimental techniques are available to investigate the parameters involved in elastohydrodynamic lubrication; film thickness using optical, electrical resistance or capacitance techniques; traction using disk machines; pressure using micro-transducers; and temperature using infra-red emission.

Ehl provides relatively thick films, low friction, and results in low wear rates and high fatigue lives. It is the failure of elastohydrodynamics which is frequently of concern to the engineer. Oil starvation at start-up, rough surface lubrication, or the effects of oil contamination are all situations where normal ehl does not occur and present a problem. It is the effects of particulate

¹ It should be noted that some of the nomenclature here differs from that of Hamrock and Dowson, the definitions of a , b , and E^* correspond with those of Johnson (1985).

lubricant contamination in concentrated contacts, which normally operate under ehl, which are of concern in this study.

2.2. Effect of Debris and Dents on EHL - a Literature Review

The majority of work carried out on the effects of lubricant debris has been based around the prediction and measurement of lubricant films generated by indented contacts.

Cheng and Bali (1980) solved the ehl equations for a sliding lubricated contact containing an idealised dent shape. Different dent shapes were modelled; pressure peaks were found to occur at the dent shoulders, deeper dents causing greater peaks. The work was extended by Goglia et al (1984) by modelling the passage of a defect as it passes through a sliding contact. Results showed the pressure peaks at the shoulders rising and falling as the dent enters, passes through, and leaves the contact.

Jackson and Cameron (1976) pioneered the use of optical interferometry to study surface defect effects on ehl. They used artificially roughened balls in an optical ehl rig and examined the effect on the lubricant film. Local reductions in the film were noted.

Wedevan and Cusano (1979) in several papers described some fundamental research into the influence of surface dents on ehl films. Using optical interferometry, they studied the progress of dents entering the contact and their effects on the lubricating oil film. They noted that a local reduction in film thickness occurred at the leading edge of a dent in the contact. It was suggested that this film reduction was caused by the loss of local ehl pressure generation whilst the dent passed through the inlet zone. This lower contact pressure at the leading edge acted to reduce the stress concentration at the dent shoulder. Since the trailing edge did not have this effect, the author hypothesised that it would therefore be a more likely site of fatigue failure. Decreases in film thicknesses were found to be greater for grooves (as opposed to dents) as lubricant was able to escape away from the contact causing a further drop in pressure.

Wedevan (1977) investigated the deformation of the dent within the film by investigating fringe patterns. Dents in a rolling ehl contact were found to deform less than the equivalent dry case as lubricant filled the dent cavity; as the film thickness increased the deformation was found to decrease. Increased slopes of the dent sides or the presence of raised edges was found to significantly increase local surface deformations (implying increase in local pressures).

Venner (1991) describes the 'multi-grid' numerical technique applied to the solution of the ehl equations. The technique dramatically speeds up the solution process and therefore allows an increase in the number of grid points; this increase in refinement then allowed the accurate study of surface dents and defects. He noted pressure peaks at the leading and trailing edge of the dent (the trailing edge spike being the greater). The deformation of the dent was found to occur in the inlet region, whilst little change occurred as it passed through the contact. Interestingly, the local change in film thickness was found to travel through the contact at the average speed of the moving surfaces (in the case of slip) and not at the speed of the indent itself.

Some research has been carried out to understand how the particles themselves effect the lubrication of a contact. The work has largely been investigations into the effectiveness of solid lubricants and particulate lubricant additives.

Sliney (1978) used optical ehl to investigate the behaviour of particles in the region of sliding concentrated contacts. He noted that materials with a high degree of plasticity, the ability to coalesce, and the ability to adhere to the surfaces were good for forming solid lubricant films. Molybdenum disulphide formed a film in the contact under dry conditions but when lubricant

was introduced the adherence decreased and the particles remained trapped in the inlet region. Silicon carbide particles were found to fracture and abrade the surfaces as they pass through the contact.

Wan and Spikes (1987) carried out some enlightening work into the behaviour of solid particles in an ehl contact. An optical ehl rig was used in conjunction with contaminated lubricants. High speed cine photography was used to capture the behaviour of various debris materials in a rolling contact. It was found that quantities of all the tested particles (sized 10-53 μm) entered the contact; generally ductile materials flattened out with little effect on the oil film. It was also noted that under conditions of high sliding debris particles blocked the contact entry region causing starvation and surface abrasion.

2.3. Optical EHL and Lubricant Debris

The last two papers reviewed in the previous section describe the use of optical interferometry to study the behaviour of particles in ehl contacts (both were aimed at studying the generation of solid films from particulates in the lubricant). Replacing one of the contacting elements with a transparent material is instructive. A 'window' can be opened on the ehl contact; features which can normally only be hypothesised can actually be viewed.

It was decided to perform some simple experiments using a range of debris materials and sizes in the rolling contact of an optical ehl rig. The aim being to gather qualitative information about debris entry and deformation within a lubricated contact.

2.3.1. The Optical EHL Rig

The test rig used in these experiments was a development of the original Gohar and Cameron (1966) optical ehl apparatus. The equipment was required to be robust and simple to withstand testing with coarse debris materials. Figure 2.3 shows a schematic of the test rig (see also photograph 2.4).

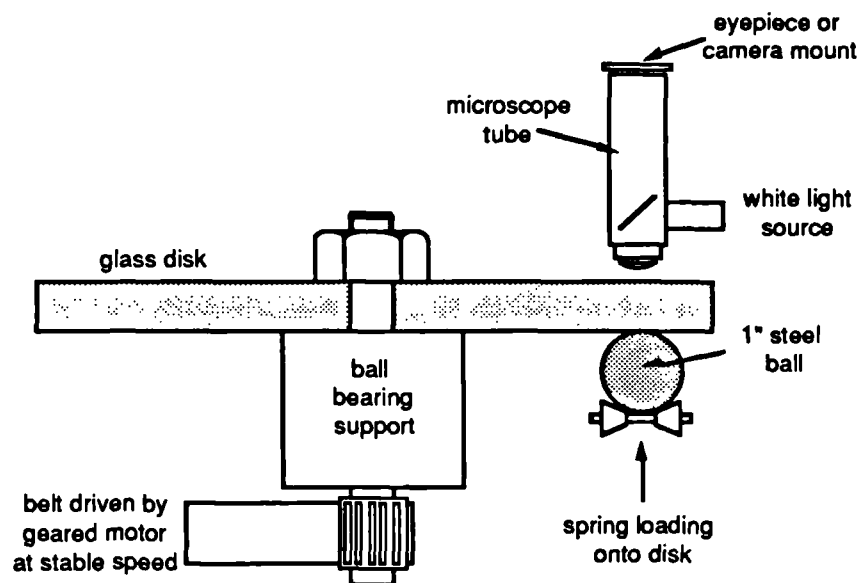


Figure 2.3. Schematic diagram of the optical elastohydrodynamic rig.

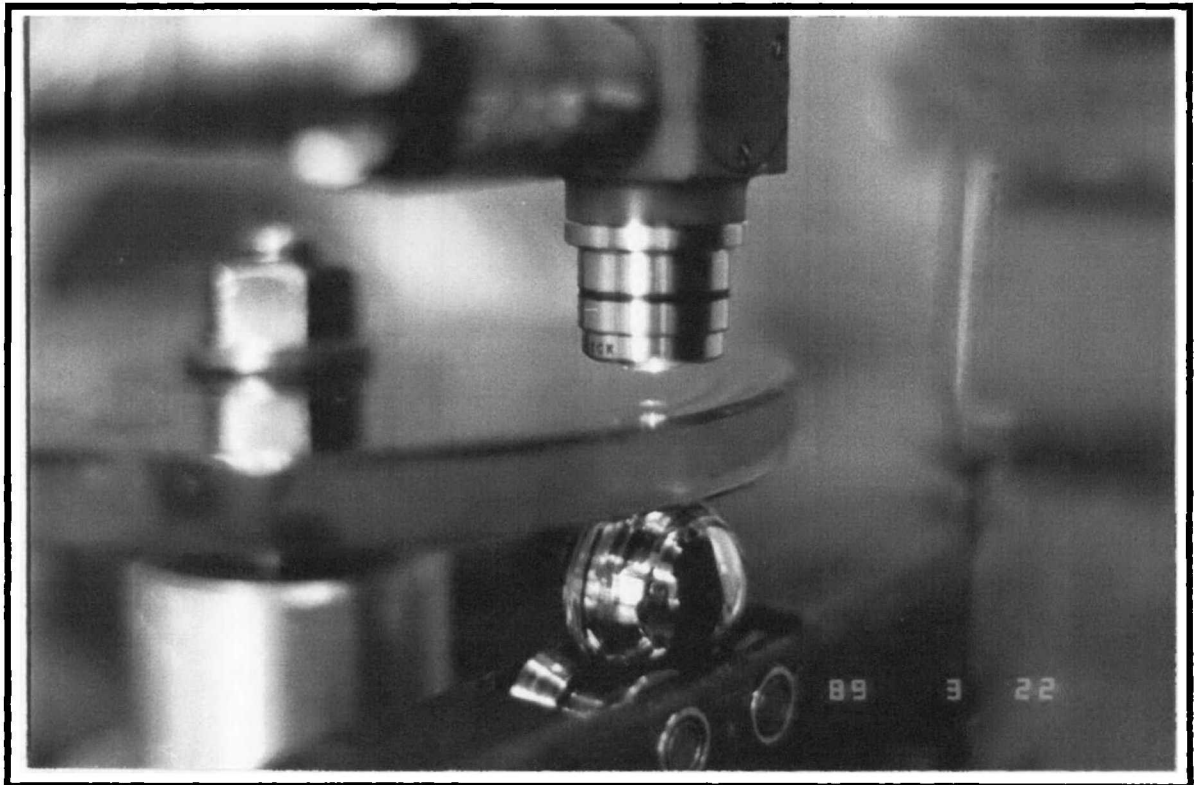


Figure 2.4. Photograph of the optical elastohydrodynamic rig. Close-up on the ball on glass disk contact.

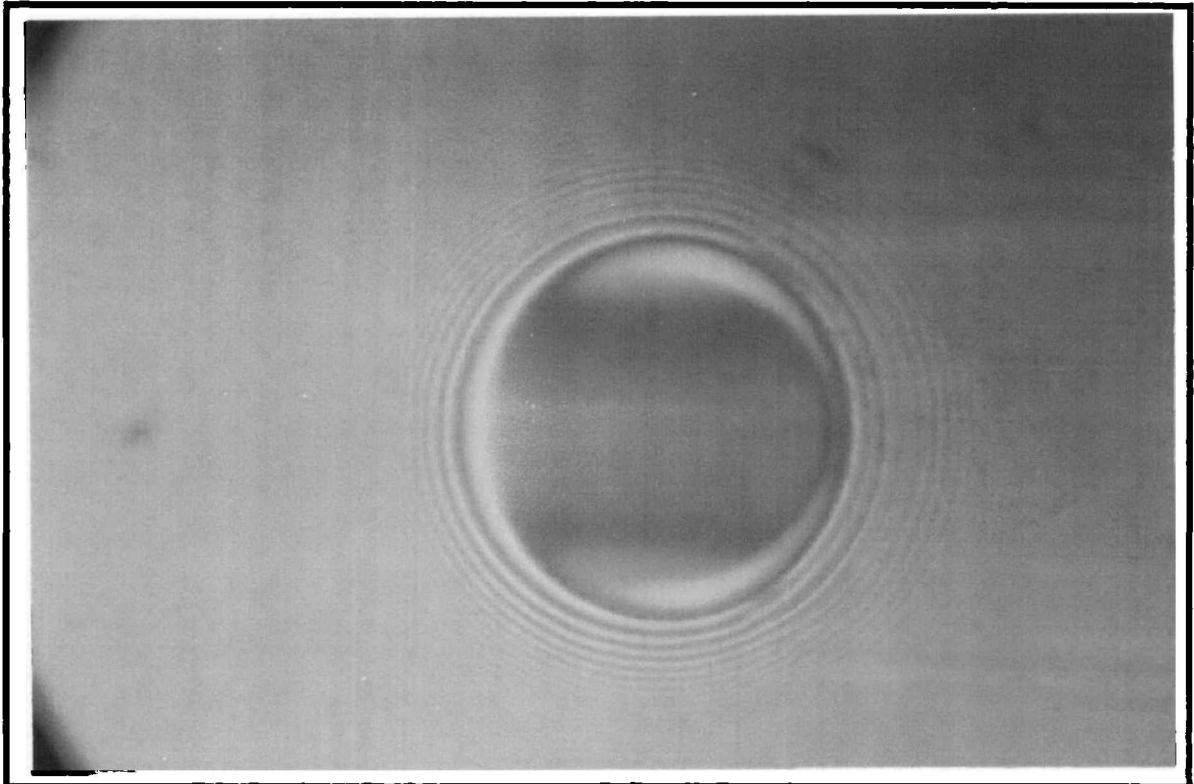


Figure 2.5. Photograph of a typical fringe pattern from the optical elastohydrodynamic rig.

A steel ball was mounted on free-moving 'cotton reels'. The support was mounted on a lever arm which was in turn spring loaded against the glass disk. The glass disk was sputtered with a 20nm layer of chromium and mounted on a vertical shaft held in a pair of ball bearings. This shaft was driven by a motor through a gear box and pulley arrangement. A purpose designed controller was used to drive the motor at low speeds with a high degree of stability.

An optical draw tube and lenses were mounted on an x-y traversing table above the contact. Light was supplied to the contact and eyepiece from a fibre optic source passed through a beam splitter. The light ray passes through the glass disk and is divided at the semi-reflective layer; part is reflected whilst the rest travels through the oil film and is reflected at the ball surface. The two reflected rays recombine and form an interference fringe pattern. The path difference is given by twice the oil film thickness. Figure 2.5 shows a typical fringe pattern.

The lubricant was mixed with debris particles and fed directly onto the disk such that flow was entrained into the contact.

2.3.2. Image Capture

Durations in the contact are very short so to capture an image required the development of new techniques. Short-duration flash photography was found to be simple, cheap, and flexible. A mercury discharge tube was used to supply a very short burst of light onto an open camera shutter. If the light is of sufficiently short duration (and the film speed high enough; ASA400 gave the best results) the particles can be captured in the contact. Generally the contaminant concentrations are sufficient (>0.5g/l) such that most photographs showed some debris captured in the contact.

To obtain a time history of debris behaviour in the contact a high speed video camera was employed. The Kodak EktaPro 1000 Motion Analyser is a black and white video camera capable of recording at 1000 frames per second. A video camera sends the pictures to a processor which are recorded on a high speed cassette. The pictures on this cassette can then be played back at 1, 2, 3, 4, or 30 frames per second and recorded on a standard VHS tape. The processor can operate at up to 6000 frames per second by dividing the image into six strips, thus each strip is 1/6000th of a second after the previous. The camera was mounted onto the microscope draw tube with high intensity lamps directed through the beam splitter. The light supply proved to be the restricting factor; it was not possible to obtain sufficient light to give a reasonable image above 2000 frames per second. Since these higher frame speeds were not practical the rolling element speeds had to be kept low (0.05-0.1m/s).

2.3.3. Calibration of the Optical EHL Rig

For film thickness measurements it is necessary to calibrate the colour of the observed fringes with a film thickness. Every operator's perception of colour is different so it is necessary for each user to perform a calibration.

Wedevan (1970) gives a technique for the calibration of the optical ehl rig. The ball is rested on the glass disk but no load applied; the radius of each fringe colour, r_f is recorded. The separation, h_c between the ball and disk may be determined by a simple geometrical calculation;

$$h_c = \frac{r_f^2}{2Rn} \quad (2.6)$$

where n is the refractive index of the lubricant, and R is the ball radius². Table 2.2 is the author's calibration chart for the optical rig used in these experiments.

Fringe order	Colour	Corresponding Separation, μm
1st	Yellow	0.19
1st	Red	0.27
1st	Blue	0.35
1st	Green	0.42
2nd	Yellow	0.47
2nd	Red	0.56
2nd	Blue	0.61
2nd	Green	0.65

Table 2.2. Calibration table used for optical ehl experiments.

The separation is an optical separation, to get the true separation the value must be divided by the refractive index of the lubricant.

The first experiment performed used this calibration to study the change in film thickness as the speed was increased. The central film thickness was measured using the colour calibration chart for the central fringe. The ball rolling speed was then increased and the measurement repeated. Figure 2.6 shows a plot of these experimental film thickness measurements against the predicted values from the Hamrock and Dowson relation (see §2.1.2 above).

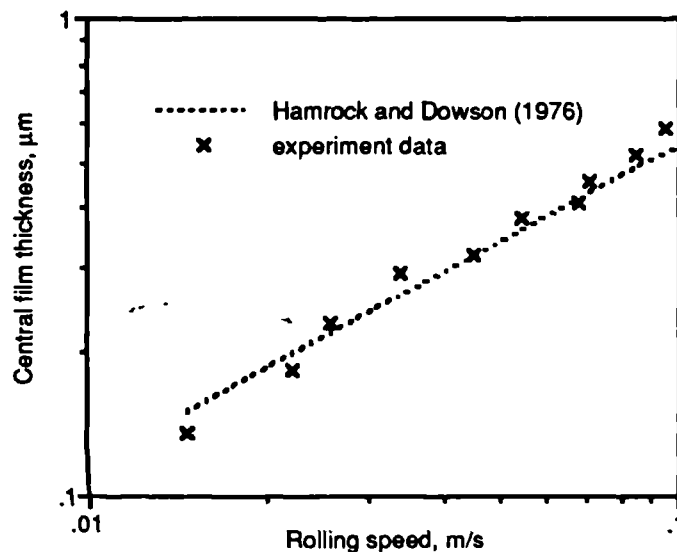


Figure 2.6. Variation of film thickness with rolling speed, optical ehl results compared with Hamrock and Dowson's relation.

² In practice it is difficult to achieve a zero load between the ball and disk and some elastic displacement occurs. To overcome this several fringe radii are recorded at different low loads and projected back to a zero load (see Wedevan 1970 for full details).

2.4. Debris in the Inlet Region and Contact

Various debris materials, with a range of hardness and fracture toughness were tested; the lubricant used was Shell Turbo T68 (Appendix 1 gives full details). The powders were mixed thoroughly with the test lubricant to a concentration of 1g/l. The mixture was fed directly onto the glass disk and entrained into the contact. Speeds were maintained at around 0.03m/s to allow for clear images to be captured. The high speed video and short duration flash photography apparatus were used to view the behaviour of particles in the inlet.

Fluid forces build up in the inlet region and smaller particles tend to be swept around the sides of the contact. This shedding process appeared to occur some 800-1200 μ m from the ehl contact centre. Particles on a central flow line or those large enough to be trapped between the closing surfaces pass into the contact. Figure 2.7 shows the mechanism.

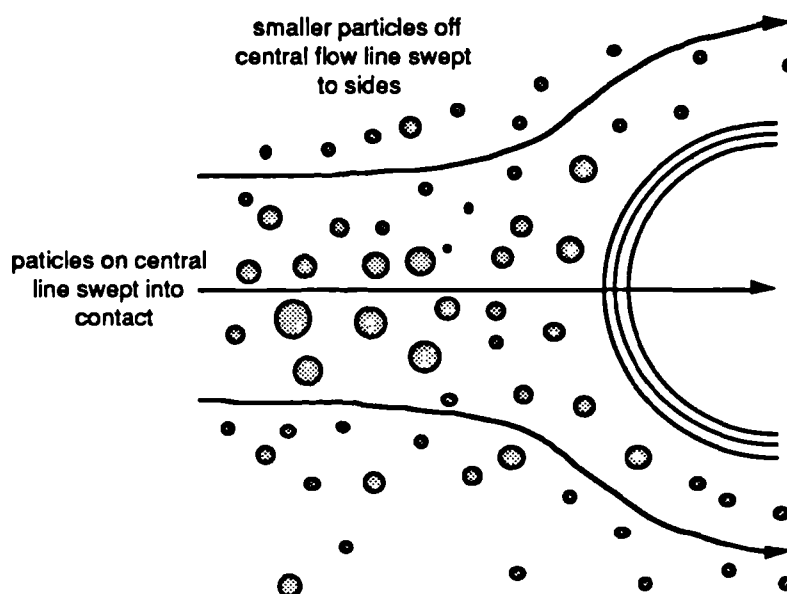


Figure 2.7. Schematic diagram of the flow of particles around a rolling ehl contact.

Ductile materials, like copper, cast iron, and low carbon steel, were flattened out in the inlet region. The platelets enter the contact. Figure 2.8 shows what were originally 30 μ m copper spheres which have been flattened out and are entering the contact. The flattened copper disks tended to adhere to the ball surface and re-enter the contact after a ball revolution. Iron and steel particles were swept out of the contact region. Examination of the overrolled debris showed the particles had been rolled into elongated flat disks with slightly split edges.

Brittle ceramic particles, like glass ballotini spheres, were fractured readily in the inlet region. 60 μ m glass micro-spheres fractured at a position approximately 1200 μ m from the contact centre. A calculation of the deflected shape of the ball (using the relation supplied by Cameron 1976) showed that at this location the separation of the surfaces was 55 μ m. The spheres are therefore fractured almost as soon as they come under load. Figure 2.9 shows the glass fragments being fragmented in the inlet region. The bright spots of light to the left of the frame are the undamaged particles. The fracture fragments are then swept around the sides of the contact; or drawn into the contact. High speed video photography showed particularly good images of this fracture process and the motion of the fragments. Figure 2.10 shows the glass spheres before and after fracture.

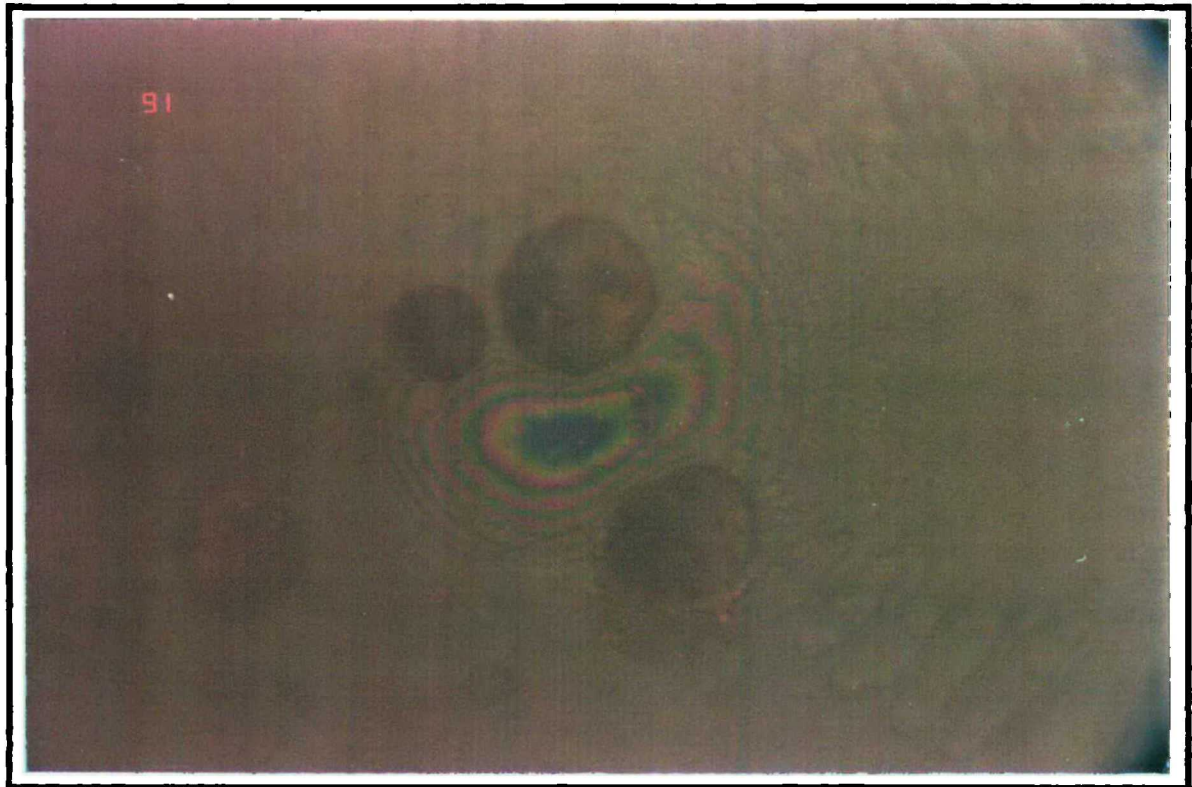


Figure 2.8. Originally 30 μ m copper spheres in a rolling optical ehl contact (magnification x100, entry direction left to right).

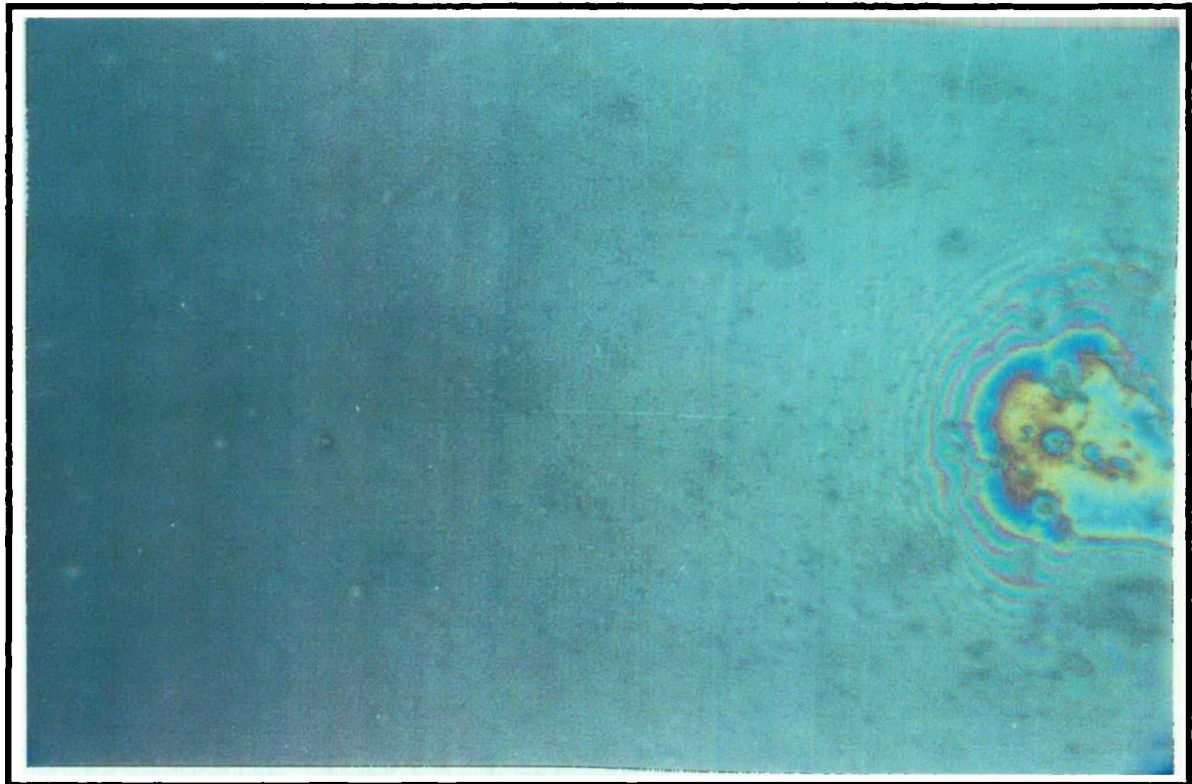


Figure 2.9. Originally 5-60 μ m glass micro-spheres in a rolling optical ehl contact. Contact is off centre to the right of the frame (magnification x100, entry direction left to right).

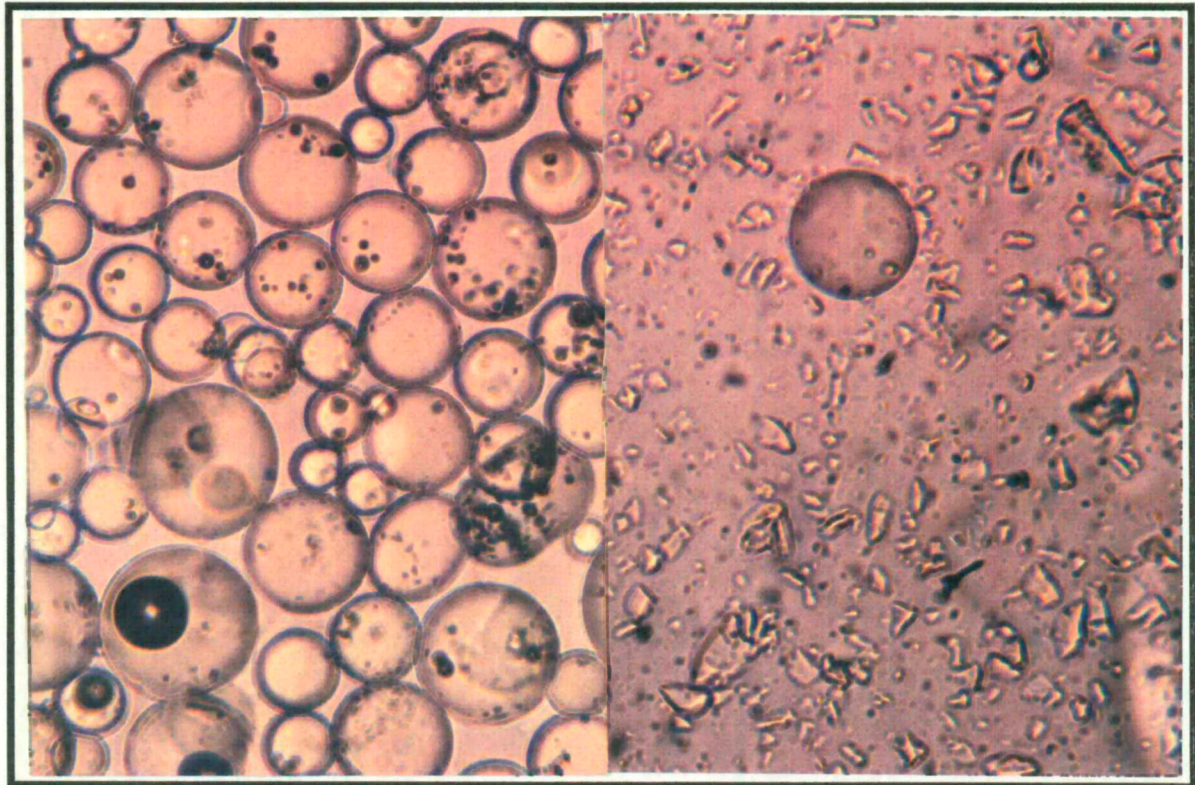


Figure 2.10. 5-60 μ m glass micro-spheres before and after overrolling in an optical ehl contact (magnification x300).

Tough ceramic particles, like silicon carbide or alumina, fracture much further into the inlet region. The fragments tended to be large and pass through the contact in a mass (causing some fracture of the glass disk). Much less dispersion of the fragments occurred and few were swept around the sides.

2.4.1. The Deformation or Fracture of Debris Particles

The size of the particles used ranged from 5-75 μ m, typically lubricant film thicknesses were less than one micron. Some considerable geometrical accommodation must take place for these particles to be overrolled. The experiments have shown that three mechanisms of deformation occur depending on the materials properties and size of the debris particles.

- (a) Ductile particles plastically deform into a platelet, contacting surfaces elastically or plastically deform depending on the relative hardnesses.
- (b) Ceramic materials fracture, contacting surfaces elastically or plastically deform depending on the debris material toughness. Fragments enter the contact and surfaces plastically deform around them.
- (c) Small size ceramic particles remain undamaged and contacting surfaces plastically deform around them.

Figures 2.11, 2.12, 2.13, and 2.14 show the debris deformation and entrainment mechanisms which occur when the various debris particles enter a contact.

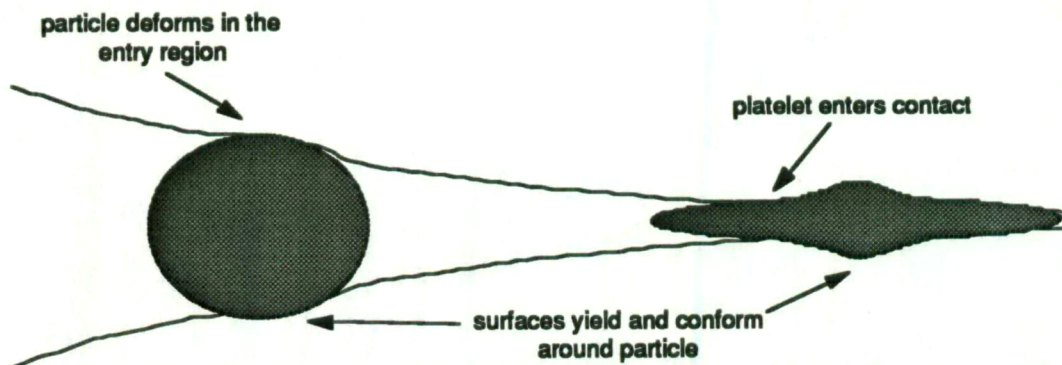


Figure 2.11. Schematic diagram of the process by which ductile materials enter a rolling ehl contact.

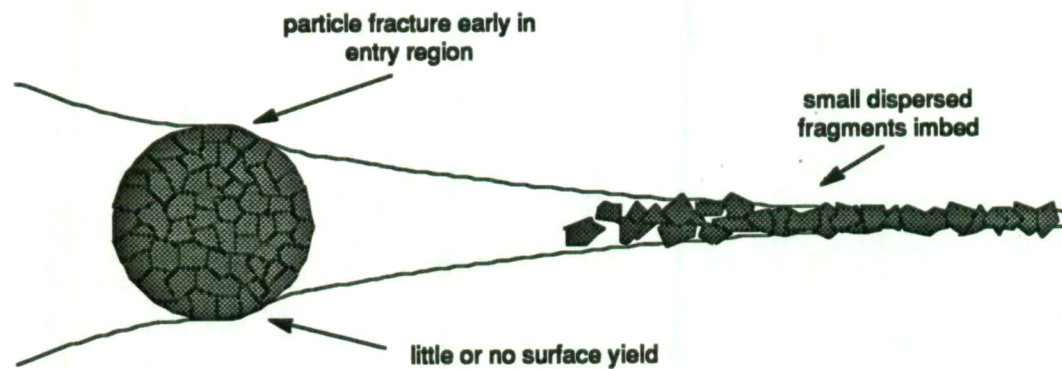


Figure 2.12. Schematic diagram of the process by which brittle ceramic materials enter a rolling ehl contact.

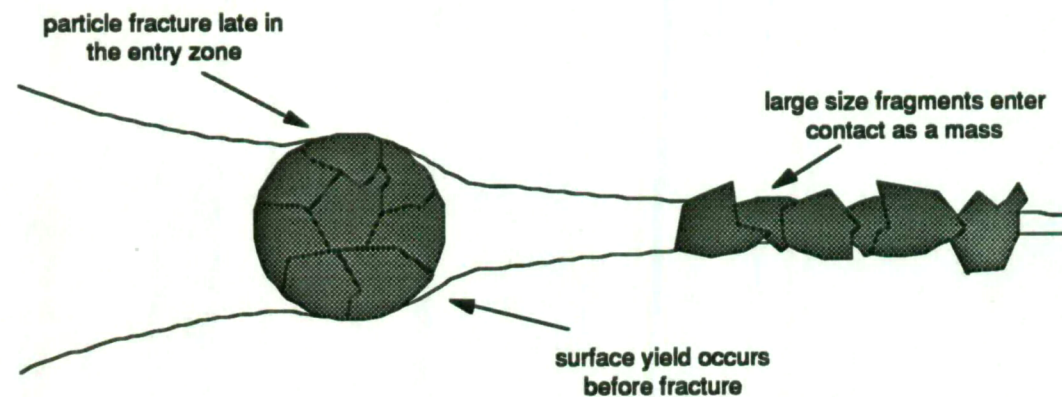


Figure 2.13. Schematic diagram of the process by which tough ceramic materials enter a rolling ehl contact.

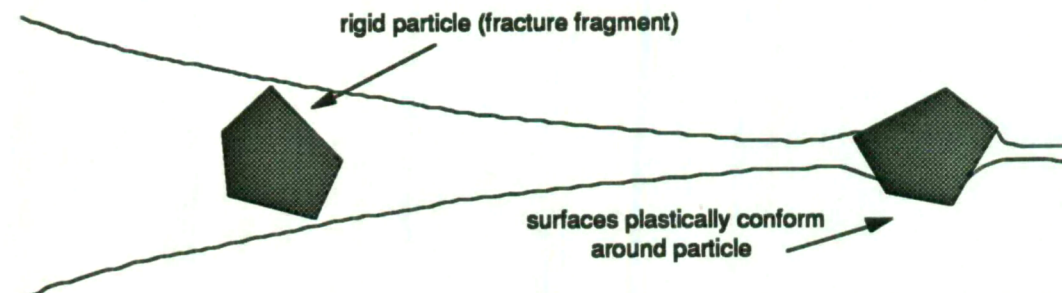


Figure 2.14. Schematic diagram of the process by which small ceramic debris materials behave rigidly and enter a rolling ehl contact.

2.4.2. The Ultimate Particle Size

When brittle particles successively fracture in the entry region, it is the fragments which finally enter the contact. Since they are of irregular shape (i.e. not smooth and flat) it is not possible for the contacting surfaces to accommodate them solely by elastic deformation; the opposing surfaces plastically conform around the remaining fragments. The size of these fracture fragments is therefore of critical importance, since it largely controls the amount of surface damage caused.

Testing with $6\mu\text{m}$ silicon carbide particles showed that no fracture occurred. Figure 2.15 shows the undamaged silicon carbide particles in the contact. It is therefore evident that this size of particle is survivable in the contact and all the surface damage will be caused by this size.

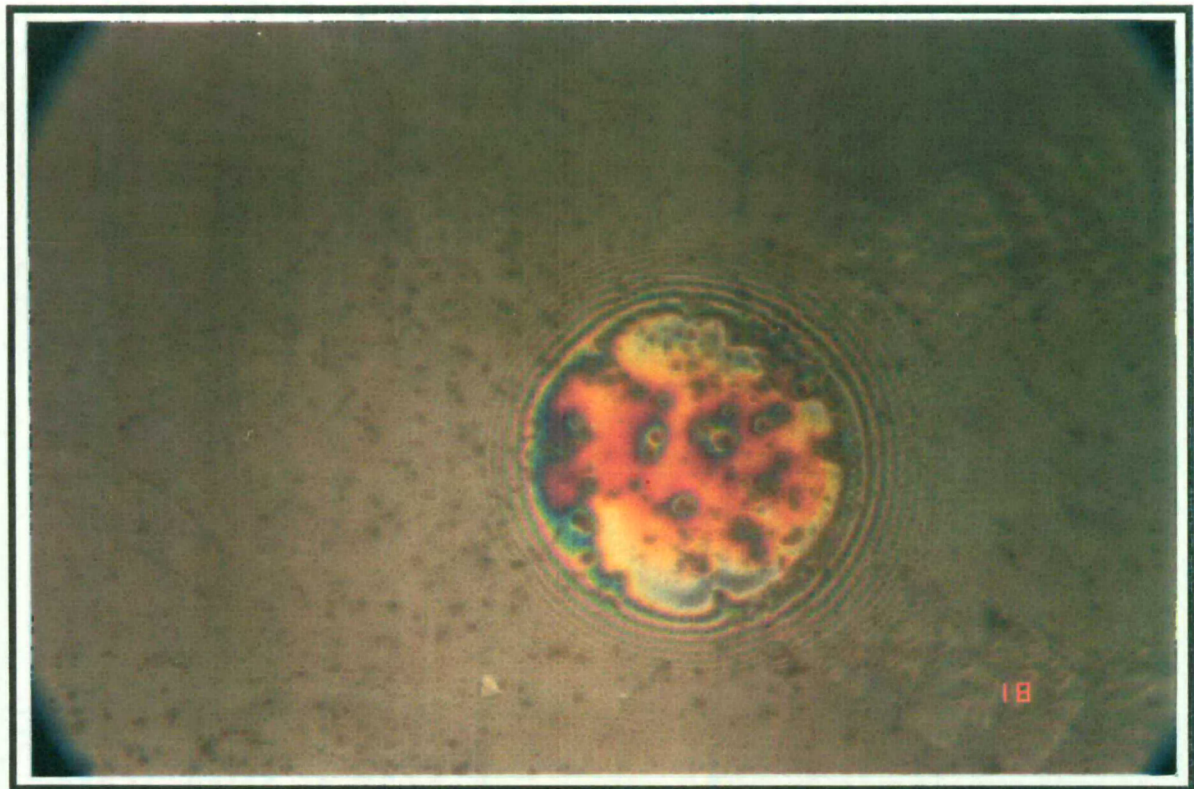


Figure 2.15. $6\mu\text{m}$ silicon carbide particles in the optical ehl contact (entry direction left to right, magnification $\times 100$).

Visual determination of the ultimate size of the fracture fragments which enter the contact is difficult. However close inspection of the photographs and the fracture fragments indicate that glass fragments of about $0.5\mu\text{m}$, alumina fragments of $3\mu\text{m}$, and silicon carbide fragments of $6\mu\text{m}$ survive the contact.

2.4.3. Ductile Flow of Ceramic Particles

Bridgman (1969) showed that normally brittle materials could be made to undergo ductile plastic flow when subjected to high hydrostatic pressures. The surrounding pressure closes any cracks which are formed and crack growth is inhibited. A hydrostatic pressure of 2.8GPa allowed glass to fail at a compressive stress of 4.3GPa ; about 25 times its strength at atmospheric pressure. The specimen underwent considerable ductile flow

The elastic stresses at the surface of concentrated contacts are close to hydrostatic. A rolling element bearing would typically be run at contact pressures in the region of 2GPa to 3GPa; plastic flow of the rolling elements or the debris material around the particle itself would raise the local contact pressure still further. It seems likely, therefore, that some ductile flow of the particles trapped in the contact may take place³.

The optical rig contact pressures are somewhat lower than those found in practice. From the photographs taken it was not possible to distinguish any ductile flow of the particles trapped in the contact; the particles are too small. However investigation of the damaged surfaces left by 1 μ m alumina particles suggest that ductile flow of ceramics could be occurring (since dents were significantly wider than the responsible particles).

2.4.4. Particle Entry into the Contact

Also of interest in the study of debris in lubricated contacts is the number and size of particles which may enter. Do certain size particles get rejected from the contact and how does the debris concentration in the bulk effect the number of particles overrolled? Simple models are presented which predict particle entry; these are used later in the thesis for the prediction of abrasive wear rates.

2.4.4.1. Size of Particles Excluded from the Contact

As a particle approaches an ehl contact it becomes trapped between the closing surfaces. If the frictional forces at the interface between the particles and surface are sufficient to draw the particle in, then it will enter the contact. Wan and Spikes (1987) presented a simple calculation for the maximum size of particle which can enter a rolling or sliding contacts; their analysis is extended here.

Figure 2.16 shows the forces on a debris particle in the inlet region of rolling and sliding contacts (neglecting the fluid forces exerted by lubricant motion). The resolved forces on the debris in the direction of entry region to a sliding contact are less than the equivalent rolling case. The particle would tend to roll in the inlet region rather than entering. Both Wan and Spikes (1987) and Sliney (1978) noted that under conditions of sliding some particles were found to roll in the inlet region without entering the contact (indeed in some instances this caused oil blockage and resulted in starvation).

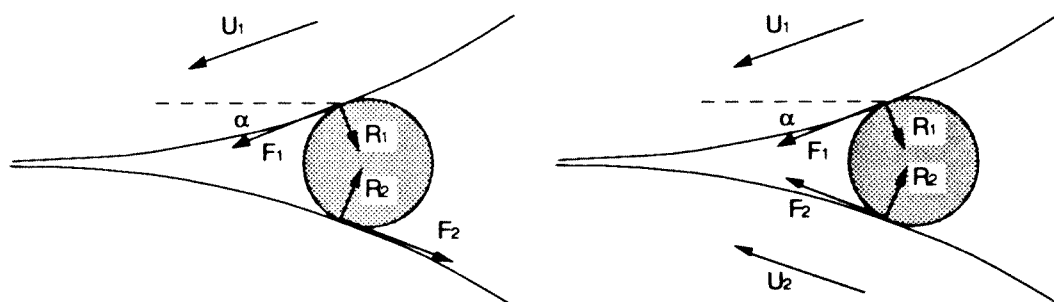


Figure 2.16. Schematic diagram of the forces on a debris particle in the inlet region of sliding (left) and rolling contacts (right).

³ Fracture of the debris particle would still take place in the inlet region where pressures are not yet developed. It is the ductile flow of the ultimate sized fragment to reach the contact which is under discussion.

Resolving the forces for the rolling case at equilibrium gives;

$$(F_1 + F_2)\cos \alpha = (R_1 + R_2)\sin \alpha \quad (2.7)$$

If μ is the coefficient of friction between the particle and the surfaces then;

$$F_1 = \mu R_1 \quad \text{and} \quad F_2 = \mu R_2 \quad (2.8)$$

substitution gives;

$$\mu = \tan \alpha \quad (2.9)$$

The angle alpha may be obtained from the geometry of the particle and rolling elements. Trigonometry gives;

$$\tan \alpha = \frac{\sqrt{rd}}{r - \frac{d}{2}} \quad (2.10)$$

and hence at equilibrium;

$$\mu = \frac{\sqrt{rd}}{r - \frac{d}{2}} \quad (2.11)$$

For a given friction coefficient and rolling element radius, the maximum particle size which may enter the contact can be determined. Figure 2.17 plots the results for a range of friction coefficients (Neale (1973) gives coefficients of 0.1 to 0.5 for boundary lubricated metals).

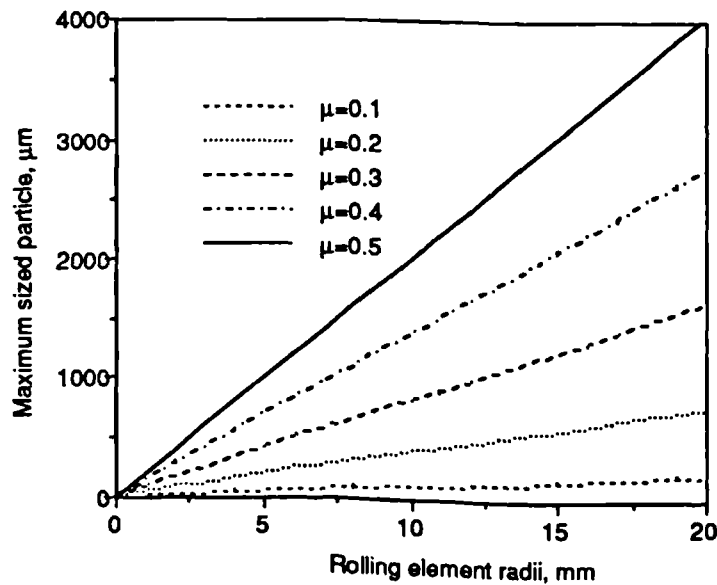


Figure 2.17. Graph of maximum particle which can enter a rolling contact against rolling element radii for a range of friction coefficients.

The analysis shows the particle size cut off at entry into a contact and how this is dependent on friction coefficient and size of the contacting elements. Wan and Spikes (1979) found that 160-200 μm glass spheres were excluded from an optical ehl contact; this result is in general agreement with the data presented here.

2.4.4.2. Number of Particles Entering the Contact

If it is assumed that the particles enter the contact as a direct ratio of their presence in the bulk of the lubricant, it is a relatively simple matter to determine the number of particles which will be present at any one instant.

The debris particles imbed in the surfaces so the volume of oil and debris in the contact is increased by a value equal to the volume of debris material imbedded into the surfaces. If it is assumed that the particles are spherical and do not breakdown or deform in the contact⁴ (i.e. the particles are similar in behaviour to fracture fragments), then the volume of oil and debris in a circular point contact is given by;

$$V_{\text{contact}} = \pi a^2 h_c + n \left(\frac{\pi d^3}{6} - \frac{\pi d^2 h_c}{4} \right) \quad (2.12)$$

where n is the number of particles in the contact, d is the diameter of a particle, and h_c is the central film thickness. The mass of debris in the contact is then given by;

$$M_{\text{debris}} = x V_{\text{contact}} \quad (2.13)$$

where x is the bulk concentration (in g/l) other units being in SI. The mass of a single spherical debris particle is given by;

$$m_{\text{particle}} = \rho \frac{\pi d^3}{6} \quad (2.14)$$

The number of debris particles in the contact is therefore the division of the last two relations. This gives;

$$n = \frac{6 x a^2 h_c}{d^3 \rho_d \left(1 - \frac{x}{\rho_d} + \frac{3 x h_c}{2 d \rho_d} \right)} \quad (2.15)$$

The graph included as Figure 2.18 is a plot of this relationship for the conditions in the optical ehl rig. The data points were determined by increasing the concentration of debris in the bulk lubricant and counting the number of particles present in photographic stills. The debris chosen was 6 μm silicon carbide particles since these were found not to fracture in the inlet region.

⁴ The calculation is more complex for ductile debris particles, where an estimate for the size and shape of the deformed platelet is required.

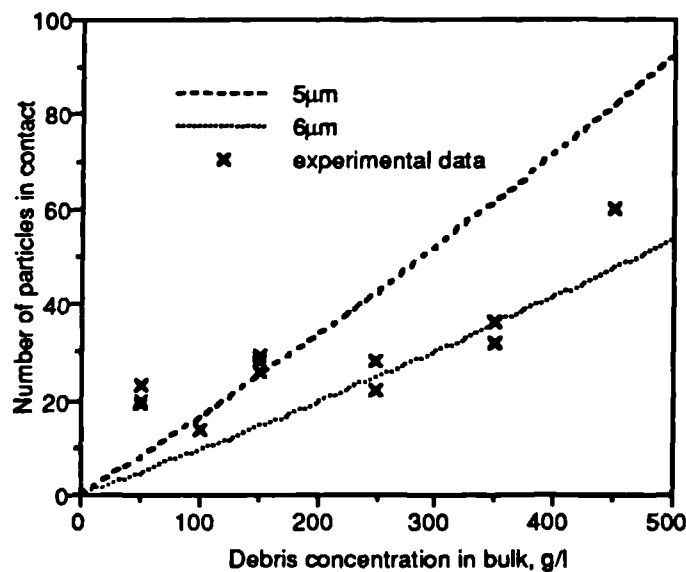


Figure 2.18. Graph of the number of silicon carbide particles ($5\mu\text{m}$ and $6\mu\text{m}$) in an optical ehl contact against the concentration in the bulk lubricant. Experimental data from optical ehl experiments.

Two concentration lines are plotted; for a debris particle size of $5\mu\text{m}$ and $6\mu\text{m}$. The number of particles is very dependent on particle size (raised to the third power in Equation 6.22). The silicon carbide particles were sized somewhere in the range 5 to 6 microns. The relation is calculated for spherical particles (which have a small volume for a given particle size); if the calculation is repeated for a more realistic cubic shape the number of particles is significantly less (by a factor of approximately $\pi/6$).

The relation gives results of the same order as the experimental data; it appears to under predict the number of particles in the contact at low debris concentrations.

2.4.5. Particle Entry into a Real EHL Contact

The optical ehl experiments and the simple models presented here are idealisations. In a realistic concentrated contact other factors will affect the entry of particles.

Fluid forces in the inlet region appear to sweep away small particles. Small particles may be less likely to enter than large particles which become trapped between the closing surfaces as the fluid forces build up. The relation provided makes no attempt to include a lower size threshold for particle entry.

The osculation of the contacting elements and the resulting elliptical contact patch may affect the particle entry. In a deep groove ball bearing, where the contact width to length ratio is of the order of 10, it is easy to visualise most of the particles becoming entrained. The process by which they become swept around the sides may be less important.

The imbedment of hard particles and the flattening of soft particles will affect their entrainment. The shape of the particle is also of importance; a sharp particle will imbed easier and be drawn into the contact.

The analyses presented here are useful as indications of how particles may enter a rolling contact. The two models are used later in the study in the prediction of rolling contact abrasive wear, where it is essential to quantify the entry of particles into a contact.

2.5. Effects on the Fluid Film

It has been established that particles may freely enter an ehl contact. It is important therefore to understand how they will effect the operation of the contact.

A simple calculation of the 'load bearing' capacity of a debris particle demonstrates that it is not possible for the particles to push the contacting elements apart and disrupt the lubricant film. If the particles have a deformed diameter, d and the surfaces conform around them, and it is assumed that the surfaces have reached a state of full plasticity. At this point the contact pressure is approximately equal to the hardness, H , of the counterfaces (the state of full plasticity is discussed more fully in Chapter 4). The load supported by n debris particles is then:

$$P = nH \frac{\pi d^2}{4} \quad (2.16)$$

For brittle materials the size of each fracture fragment is small and therefore the load supported is very low. For $6\mu\text{m}$ silicon carbide, which undergoes little deformation, at a concentration of 1g/l the load supported by the debris particles would be approximately 0.02N . Thus, only 0.002% of the contact is debris to metal rather than metal to metal contact.

In the optical experiments performed, at no time was the film distorted by the particles. Even at a very high debris concentration, 400g/l the percentage of debris to metal contact was less than 1% . Figure 2.19 displays the results of an experiment to demonstrate this; the film thickness in the optical ehl rig was measured for a range of debris concentrations. The film thickness does not appear to depend on the presence of particles in the lubricant.

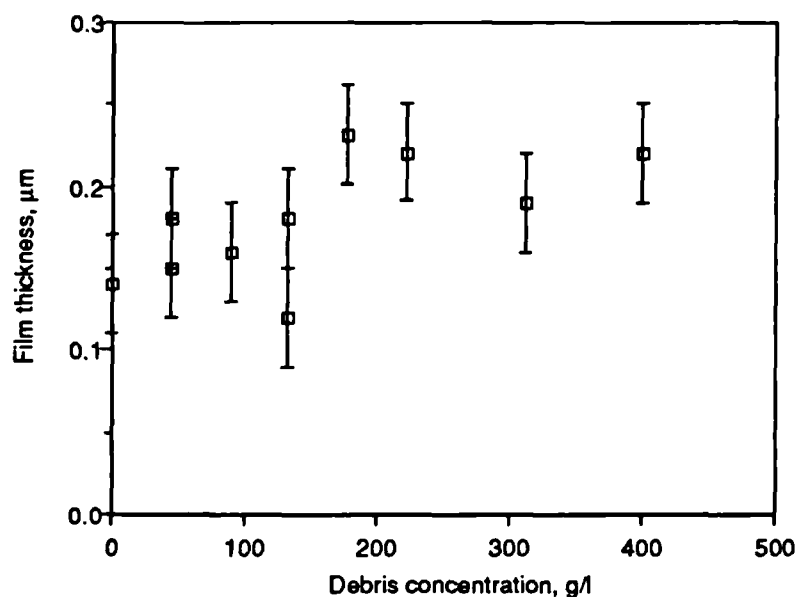


Figure 2.19. Graph of the measured film thickness of an optical ehl contact with a lubricant containing various concentrations of $6\mu\text{m}$ silicon carbide debris.

A similar result would be expected using ductile debris materials. Since the load to squash a particle is very small (compared with the applied contact load), the debris would be flattened out and the rolling elements conform around it. The whole process of debris indentation in concentrated contacts is geometry controlled. Increasing the load does not cause further indentation of the debris particles. It may cause an increase in the contact area and therefore an increase in the number of particles entrained; but the depth of indentation of each particle is independent of applied loads, at realistic operating conditions.

This mechanism is very important when one considers abrasive wear by debris particles; the wear rate is geometry controlled and not, like other forms of wear, load controlled. This is discussed further in later chapters.

It is important not to misinterpret the effects of the bulk of the lubricant film surviving. The purpose of the lubricant film is to separate the contacting components. The film is still intact but surface contact occurs by the debris particles penetrating the film. The lubrication, in this instance, is therefore not performing adequately.

2.6. Reflections on Surface Damage

The mechanisms by which the debris materials behave in the inlet region to a contact have a significant influence on the nature of the surface damage generated.

Ductile debris particles are rolled out in the contact and leave shallow rounded smooth dents. The process implies that the damage is dependent on the particle size and hardness. These parameters will control the flattening of the particle and hence the plastic conformity of the counterfaces.

Tough ceramic particles fracture late in the entry zone and the large size fragments imbed as a mass. They leave clusters of deep dents with steep sides and raised edges. The damage is controlled by the particle size and toughness. These parameters indicate at what point in the contact entry region the particle break and how large the fragments are.

Brittle ceramic materials fracture readily in the inlet region and the small fragments disperse, leaving a surface crazing of shallow fine dents. The damage is controlled by the fracture toughness alone; the size of the debris is less relevant since the particle becomes crushed down to fine fragments (although, of course, a larger particle will produce many more fragments which may then be overrolled).

2.7. Conclusions

The work described in this chapter was aimed at determining the behaviour of various material debris particles in and around an ehl contact. Replacing one of the contacting elements by glass, in an optical ehl rig, provides a technique for 'looking in' to the contact. High speed video and short duration flash photography were used to freeze the motion of the particles in a contact. The following points are concluded from this part of the study;

- (i) Ductile materials were found to flatten in the entry region and the deformed platelet entered the contact.
- (ii) Brittle materials were found to fracture in the inlet region and the fragments, whose size depends on the toughness of the material, imbedded in the contact. The size of crushed

Arizona dust fragments were estimated as less than $1\mu\text{m}$; whereas $3\mu\text{m}$ alumina and $6\mu\text{m}$ silicon carbide particles were found to survive undamaged.

- (iii) Small debris particles can be swept around the sides of this optical rig contact.
- (iv) Large particles may not be entrained into the contact, since remote from the contact frictional forces in the direction of entry are small.
- (v) The debris particles did little to upset the ehl film. Loads supported by the particles were small compared with contact loads. Typically only about 0.05% of the load would be supported by debris particles in a 1g/l concentration.
- (vi) The number of particles which entered an optical ehl contact were approximately in proportion to the debris concentration in the bulk.

The information gathered in this chapter is used in later sections of this thesis. The mechanism of debris deformation in the inlet region has a significant effect on the resulting surface damage. And, the entry and behaviour of the particles is relevant to the understanding of three body abrasive wear.



CHAPTER 3

DEBRIS AND SURFACE DAMAGE

3.1. Introduction

A major step in investigating the relationship between lubricant contamination and bearing life is, by the nature of the failure processes, an assessment of the surface damage generated by the overrolling of debris particles. A given lubricant will contain a mix of debris particles of various sizes and materials. The use of this lubricant in a rolling element bearing will result in some form of element damage which is related to the contaminating particles.

The purpose of this part of the project has been to define the relation between debris particle characteristics and ensuing surface damage. A purpose built rig has been used to study debris damage on a pair of rollers in a realistic elastohydrodynamic contact. The data from these experiments is compared with typical dents from real applications and predictions from indentation models.

3.2. Debris Initiated Surface Damage - a Literature Review

Webster (1986a), as part of study of debris initiated cylindrical roller bearing fatigue, studied the damage caused by the overrolling of a lubricant contaminated with gear box debris. He used surface mapping equipment to produce topographical maps of the damaged surfaces⁵. Many surface maps and contour plots were generated, dents depths varied in size from 0.6 to 6 μ m, and diameters from 24 to 500 μ m (although dents of less than 5 μ m were observed but were below the resolution of the mapping system). Dents were generally shallow with slopes of ranging from 1° to 10°, shoulders were rounded and in some cases raised from the surface. The spread of data was attributed to the random nature of the debris responsible for the damage and that self generated debris from the failed specimens may also have been overrolled. A mechanism was then presented to explain how a softer particle can indent a hard bearing surface; the elastic conformity of the bearing surfaces around the particle and the restricting hydrostatic ehl film were thought to restrain plastic flow of the particle and cause denting of the counterfaces.

⁵ The specimens were cylindrical roller bearings obtained from the fatigue experiments of Bachu (1980) Lubricants were circulated through a highly loaded gearbox which generated a supply of wear debris.

In a follow up to this work on debris related fatigue of rolling bearings Hamer et al (1989) carried out studies to define this damage relationship for ductile particles. Pellets of ductile materials were squashed between hard cobalt tool steel platens, and the resulting surface damage examined. Figure 3.1, reproduced from the publication, shows talysurf profiles of four such dents.

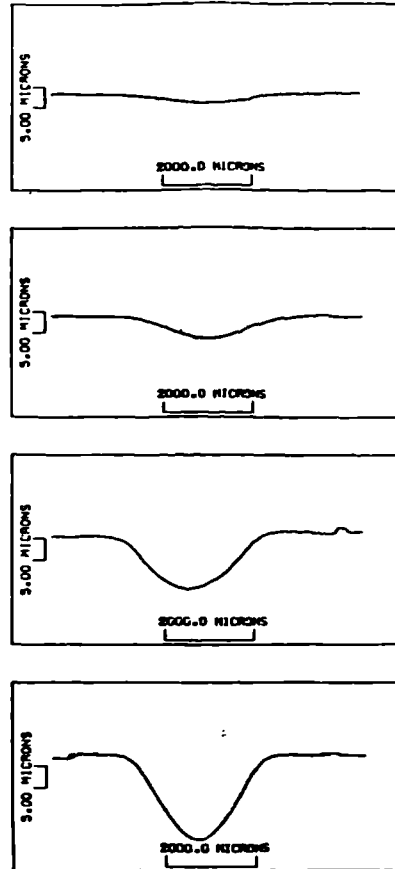


Figure 3.1. Talysurf profiles of the dents caused by the squashing of (top to bottom) aluminium, copper, mild steel, and low carbon steel. Pellet hardnesses were 304, 212, 123, and 51V respectively. Counterface hardness 1100V.

The dents were found to be very dependent on the material hardness, and soft materials could indent the much harder surfaces. The debris particles were found to work harden by some 40 to 60%.

Hamer et al (1989) developed a model to predict this behaviour where the particle was considered to be purely plastic of constant volume. The extrusion pressures were calculated by resolving forces on elemental rings; these pressures were iterated to balance the resulting elastic deflection of the counterfaces. The model was extended to include plasticity of the indented surfaces by using a pressure profile derived from Johnson's cavity model (1985); this was then iterated with an approximate plastic deflection equation. Dents generated using this method gave a good approximation to those found in the experiments. The model exhibited higher errors at the extremes of the particles size to film thickness ratio; where particle deformation energy and elastic-plastic displacements are more important.

This pellet squashing work was limited to ductile particles. This author carried out a similar series of tests using brittle materials (see Dwyer-Joyce et al 1990). The pellets fractured readily and as the load was increased successive fractures occurred. Eventually the anvils closed

around the fragments. Three brittle materials of varying fracture toughness were tested; quartz, silica, and alumina (of toughness approximately 0.5, 1, and 4 MPa $\sqrt{\text{m}}$ respectively). Figure 3.2 shows talysurf profiles of the anvil dents resulting from loading up to 150kN.

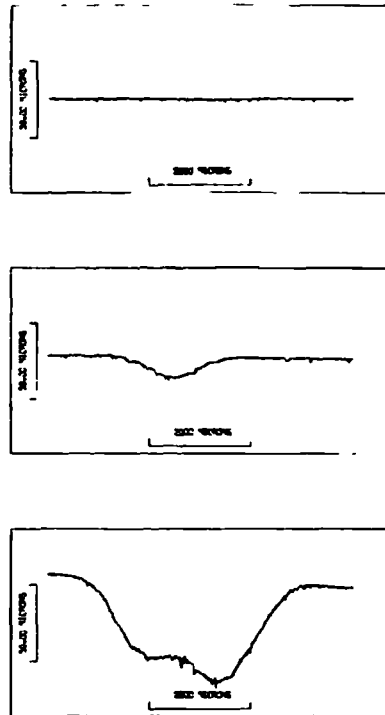


Figure 3.2. Talysurf profiles of the dents caused by the squashing of (top to bottom) quartz, silica, and alumina to a load of 10 tons.

The tougher the pellet material the deeper the resulting dent. The low toughness brittle materials show only slightly damaged surfaces where the fragments have become enclosed by the counterfaces. For the tougher materials the fragments appeared to indent the counterfaces as a mass, leaving a deeper rounded dent with a rough surface. This was thought not to be a good analogy of the debris overrolling process, and no further work was attempted.

3.3. Debris Overrolling Experiments

The work, to date, has been based on the mapping of damaged bearing surfaces or on pellet squashing experiments. In these tests a purpose built rig was used to obtain a closer analogy to the debris overrolling process. The simple contacting pair of spherical rollers provides a test cell for the overrolling of realistic size debris particles in an elastohydrodynamic contact.

3.3.1. The 'Mangle Rig'

In this apparatus debris particles in a lubricant are overrolled by a pair of rollers in an ehl contact. Crowned rollers were used for ease of alignment, and to give an elliptical contact patch similar to that found in a ball/raceway contact in a rolling element bearing. Contact pressures and speeds were chosen to be comparable to those found in a rolling element bearing.

Figure 3.3 shows a schematic diagram of the rig used in these experiments. The two cylindrical bearing steel rollers under test were held between four large nylon rollers. The nylon rollers and spindles were machined and mounted exactly so as to accurately align the test elements. A

motor with variable speed controller was used to rotate the four nylon rollers at speeds in the range 0 to 6000rpm. An arrangement of gears and belts allowed the four nylon rollers to rotate in such a way that nominally pure rolling was induced in the two contacting elements. Load was applied through a hydraulic loading stud. One of the nylon roller pairs was fixed whilst the other was free to pivot under load (as demonstrated in Figure 3.5).

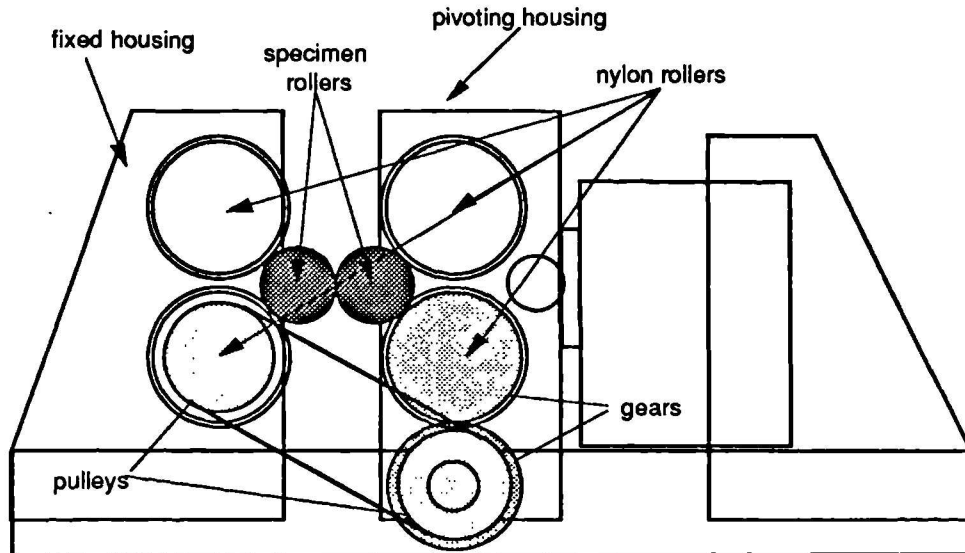


Figure 3.3. Schematic diagram of the mangle rig

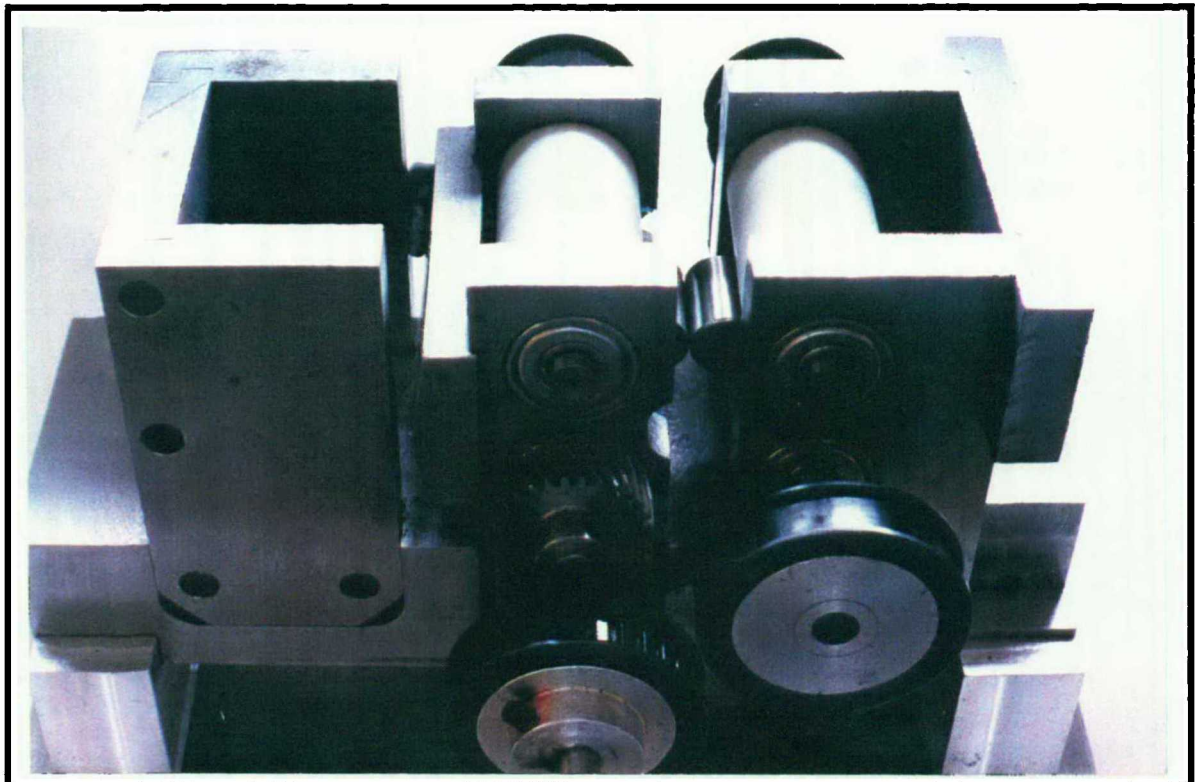


Figure 3.4. Photograph of the mangle rig (without motor and loading assembly)

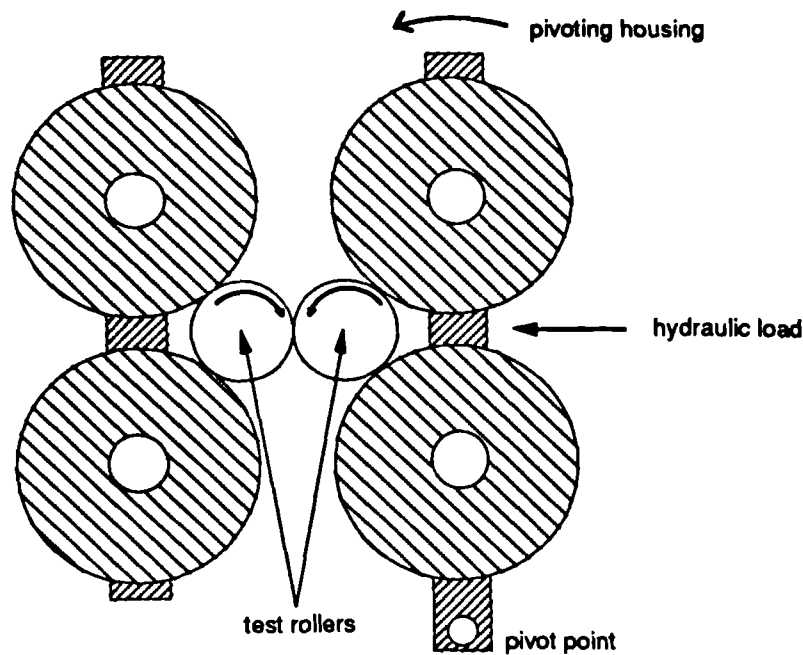


Figure 3.5. Schematic diagram of loading and pivoting mechanism.

A simple rubber faced seal assembly was fitted around the contacting rollers; this ensures that the lubricant and debris had a greater chance of being overrolled. A drain in the underside of the seal assembly was connected via neoprene tubing to a peristaltic pump for collection of overrolled lubricant and debris. Figure 3.6 shows the contacting roller pair and seal assembly and Figure 3.7 shows a schematic of the whole test apparatus. The photograph included as Figure 3.4 shows the rig (without the roller seal, motor, or loading assembly).

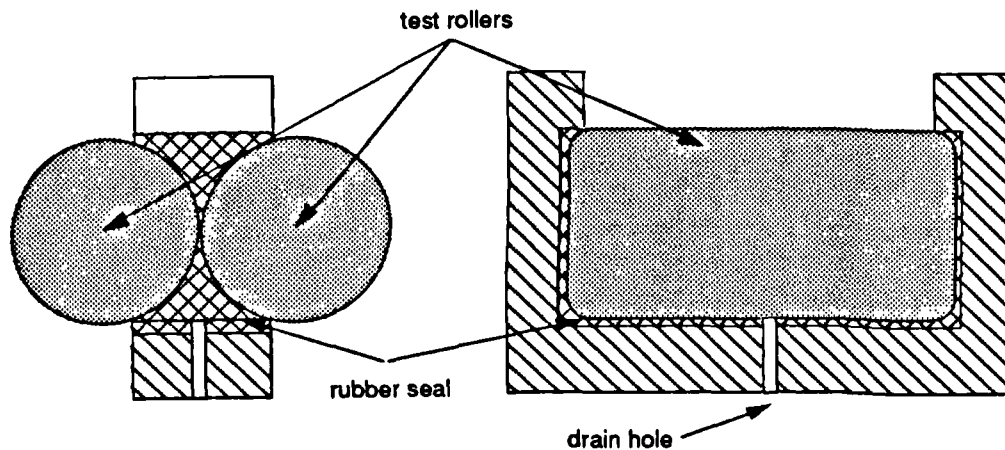


Figure 3.6. Schematic diagram of the contacting rollers and seal assembly

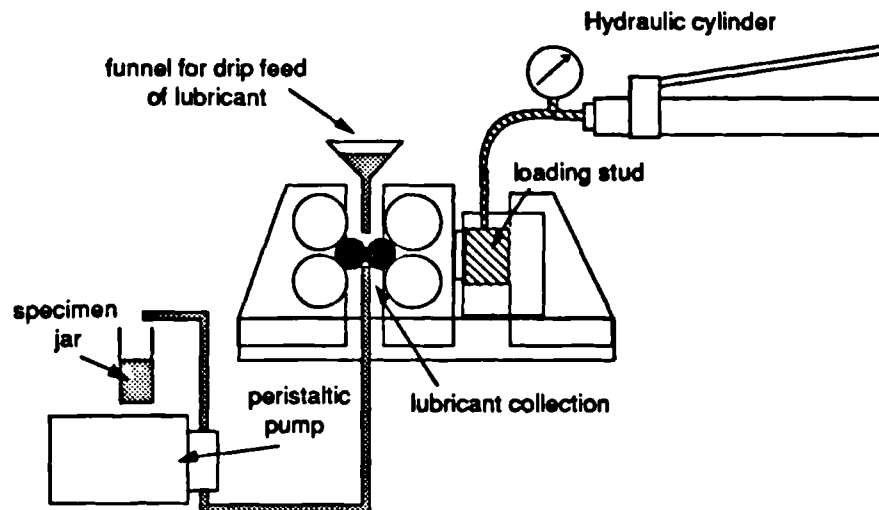


Figure 3.7. Schematic diagram of the mangle rig and test apparatus

3.3.1.1. Test Specimens

Tables 3.1 and 3.2 list the debris powders used. The materials were chosen to give a range of hardness or fracture toughness; and also because these materials are likely to be found in a practical lubrication system.

Material	Size range, μm	Particle shape	Hardness, GPa
Low Carbon Steel	0-60	irregular	1.5-2
Cast Iron	0-60	irregular	0.8-1.0
Copper	0-60	spherical	0.4

Table 3.1. Ductile debris powders used in the overrolling tests (material properties are estimates only).

Material	Size range, μm (before sieving)	Particle shape	Toughness, $\text{MPa}\sqrt{\text{m}}$
Arizona dust	0-100	sharp rough	1.5
Glass Ballotini	0-60	smooth sphere	0.7-0.8
Boron Carbide	0-60	crystalline	1.7
Alumina	0-100	sharp rough	3-4
Silicon Carbide	0-100	cubic crystalline	4-5

Table 3.2. Brittle debris powders used in the overrolling tests (material properties are estimates only).

Powders were generally obtained as a broad size distribution and were subsequently sieved. The same size range was used for each of the debris materials (58 to $63\mu\text{m}$); this range was thought to be characteristic of the size of real damaging debris found in practice and would also give a measurable scale of surface damage. Concentrations of 1g/l in Shell Turbo T68 were used. Optical microscopy was used to qualitatively define the particle shape.

The test elements were crowned rollers from cylindrical roller bearings, with a rolling radius of 7.5mm and a transverse radius of 5000mm. Appendix 1 gives further details of the test debris, lubricant, and specimens used.

3.3.1.2. Experimental Method

The debris powders were thoroughly mixed with the test lubricant. The test rollers were loaded to 2000N giving a peak contact pressure of 1.6GPa (Appendix 1 gives details of the contact conditions), and rotated at 3000rpm giving a surface speed of 2.4m/s. The lubricant and debris mix was fed directly onto the roller pair. The peristaltic pump, connected to the seal drain, was then switched on and the overrolled lubricant debris mix collected. After completion of the test the rig was thoroughly cleaned and the roller pair replaced.

3.3.1.3. Test Measurements

The surface damage on the rollers was qualitatively assessed using an optical microscope. The surface profiles were obtained using a Form Talysurf; in addition 3D maps were generated. Since a large distribution of debris dents is present a number of profiles are recorded and a range of dents measured.

The particle size distribution in the overrolled lubricant and debris mix was determined using a Malvern particle size analyser. In this apparatus laser light is shone through the sample. The amount by which the light is scattered gives a measure of the particle size distribution. Distributions of untested and overrolled debris particles were compared.

3.3.2. Ductile Particle Damage

Three ductile debris materials were tested; low carbon steel, cast iron, and copper. The nature of the surface damage on the contacting rollers is discussed in this section.

3.3.2.2. Indent Shape

Figures 3.8, 3.9, and 3.10 are 3D surface maps of the surface damage caused by the 58 to 63 μ m low carbon steel, cast iron, and copper particles. The photographs included as Figures 3.11, and 3.12 show the nature of the dents.

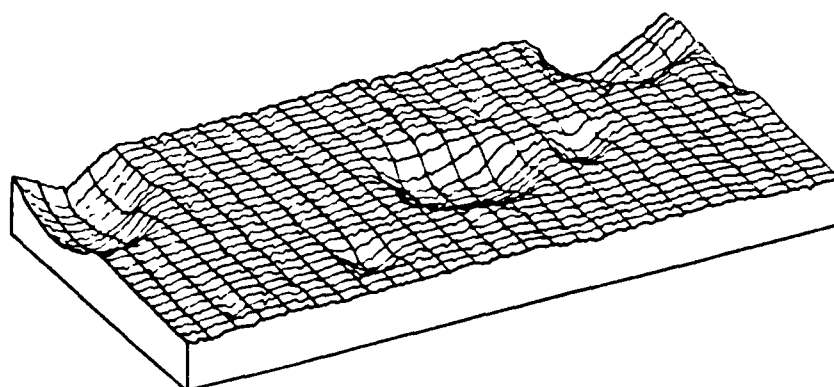


Figure 3.8. 3D surface map of roller surface damaged by low carbon steel (patch size 0.5mm x 1mm vertical magnification 25x horizontal)

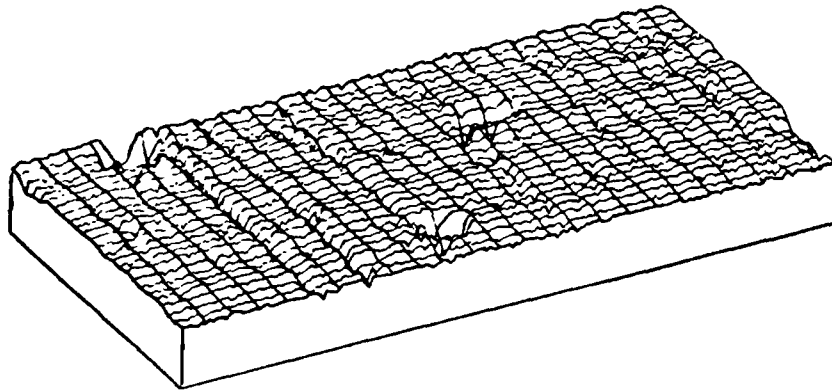


Figure 3.9. 3D surface map of roller surface damaged by cast iron (patch size 0.5mm x 1mm vertical magnification 25x horizontal)

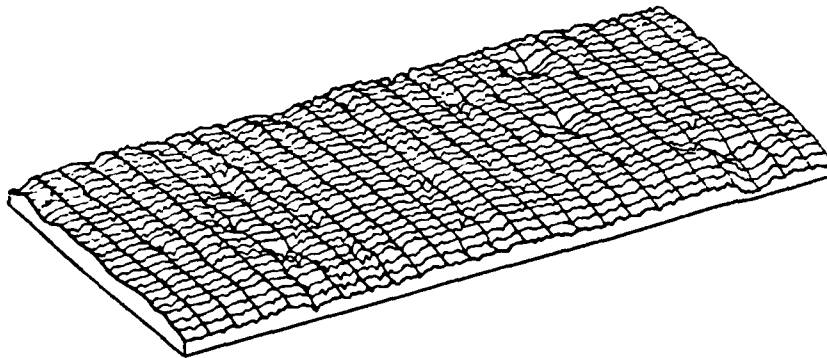


Figure 3.10. 3D surface map of roller surface damaged by copper (patch size 0.5mm x 1mm vertical magnification 25x horizontal)

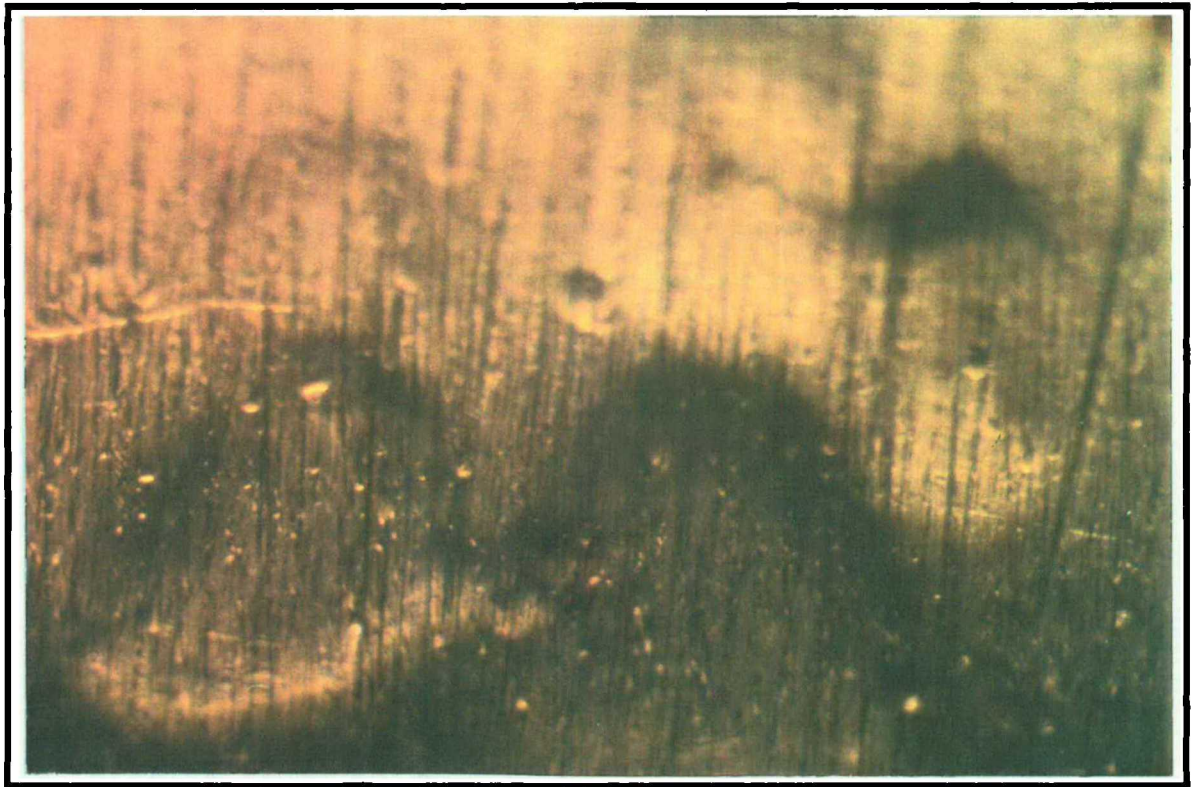


Figure 3.11. Photomicrograph of roller surface damaged by low carbon steel (magnification x300)

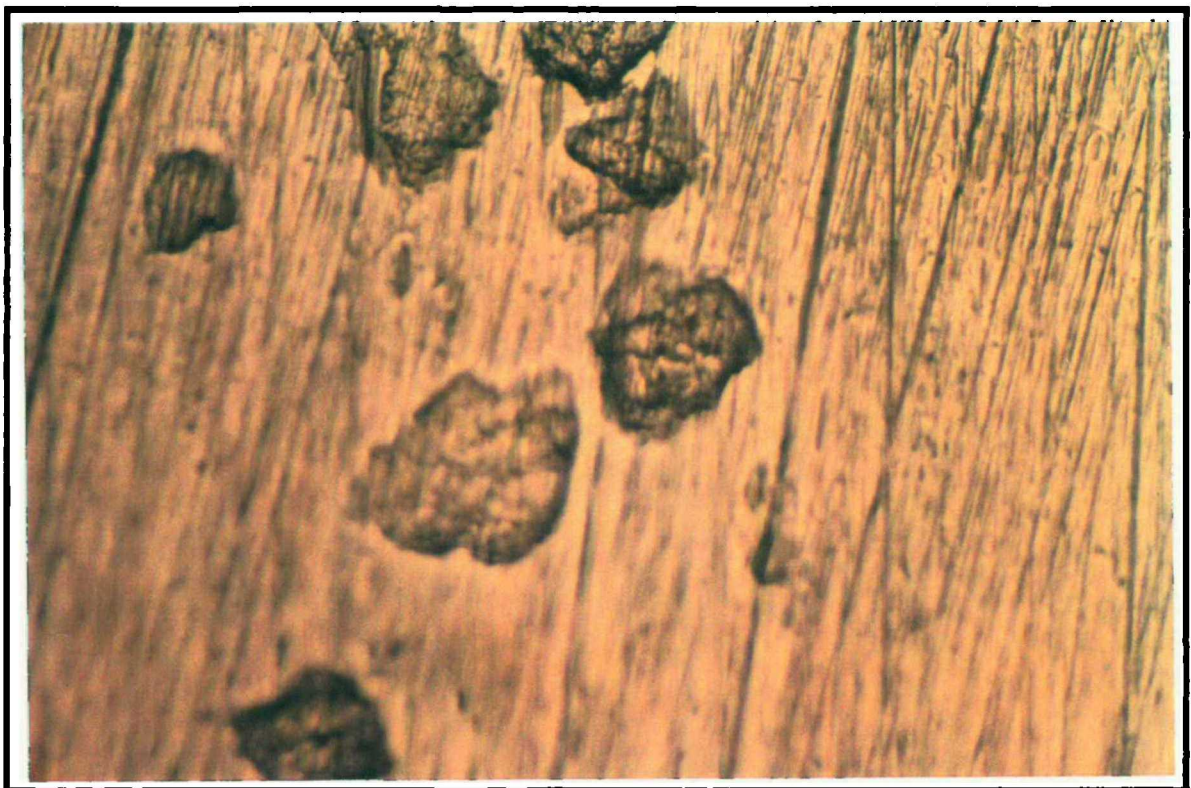


Figure 3.12. Photomicrograph of roller surface damaged by cast iron (magnification x300)

The dents are generally smooth with slightly raised shoulders. The harder materials leave deeper dents. The low carbon steel dents have the grinding marks visible; whilst those for cast iron have a rougher frosted appearance. It appears that the copper and steel particles undergo a ductile flattening process. The cast iron suffers some process involving fracture during the ductile flattening. This leaves the rounded dents and the imbedding of the fragments leaves a rough surface

3.3.2.1. The Effect of Debris Hardness

The depth of indentation is dependent on the debris hardness. Table 3.3 shows typical dent sizes caused by the overrolling of 58 to 63 μm particles of different material.

Material	Dent depth, μm	Dent width, μm	Damage Appearance
Low Carbon Steel	1.0-2.0	150-200	Rounded smooth
Cast Iron	1.5-3.0	100-150	Frosted base
Copper	0.5-1.0	100-150	Shallow rounded

Table 3.3. Damage size and appearance caused by ductile debris materials.

3.3.2.2. The Appearance and Distribution of Overrolled Particles

The debris particles are flattened during the overrolling process. The cast iron particles are somewhat more fragmented with tattered edges and evidence of small fragments. Photographs included as Figures 3.14 and 3.15 show cast iron and copper particles before and after overrolling.

The distribution of copper particles in the contaminated lubricant, before and after testing, was measured. Figure 3.13 displays the results. The equipment uses a magnetic stirrer and is therefore not suitable for analysis of ferrous debris materials.

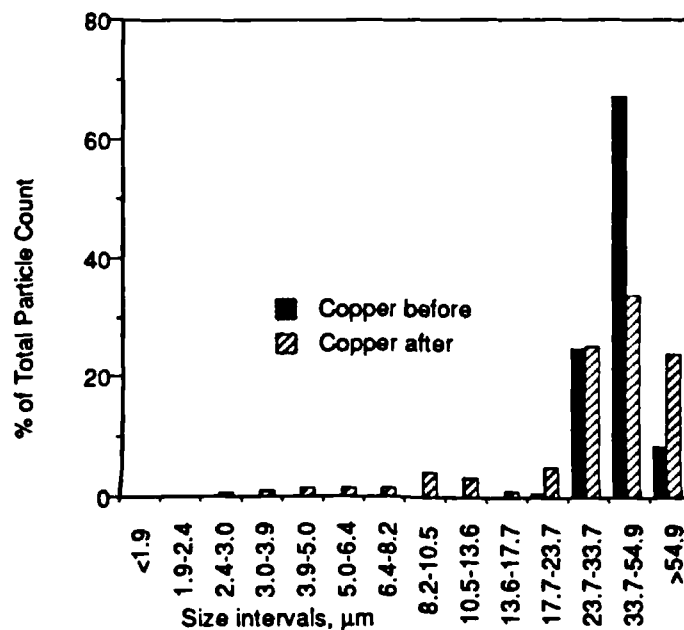


Figure 3.13. Particle size distributions for copper particles before and after overrolling

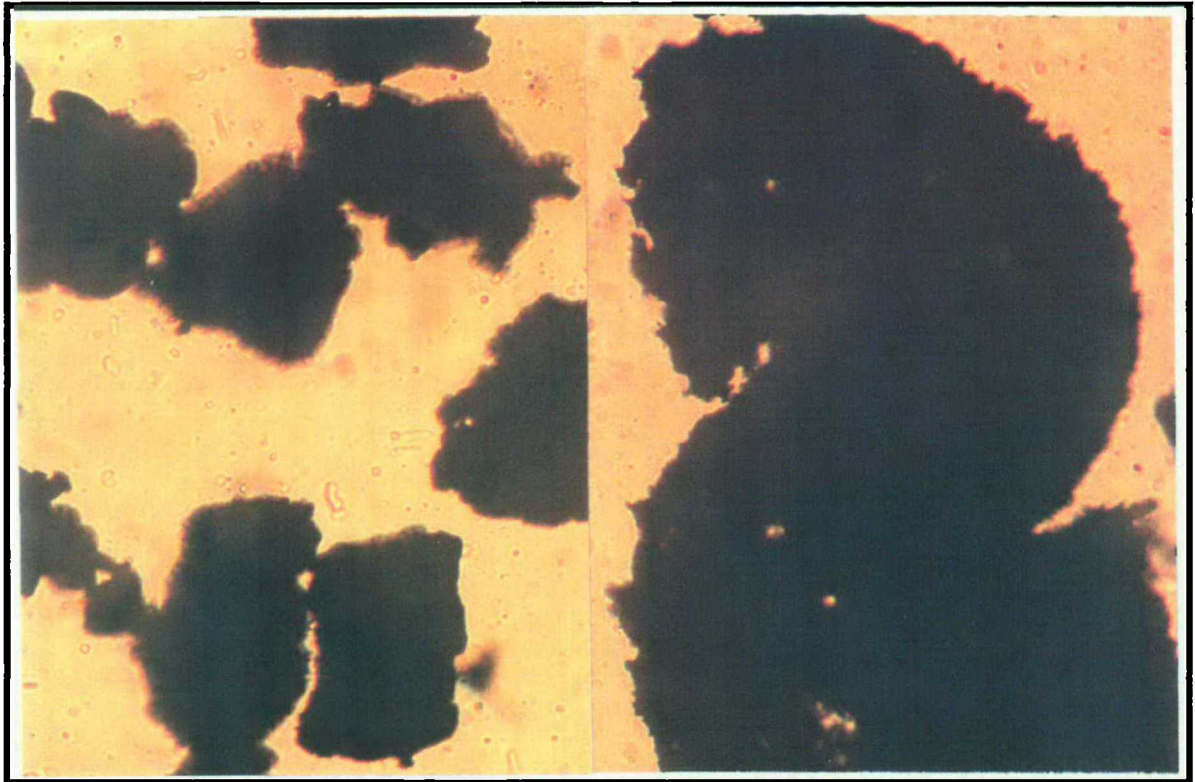


Figure 3.14. Photomicrograph of cast iron particles before and after overrolling (magnification x300)

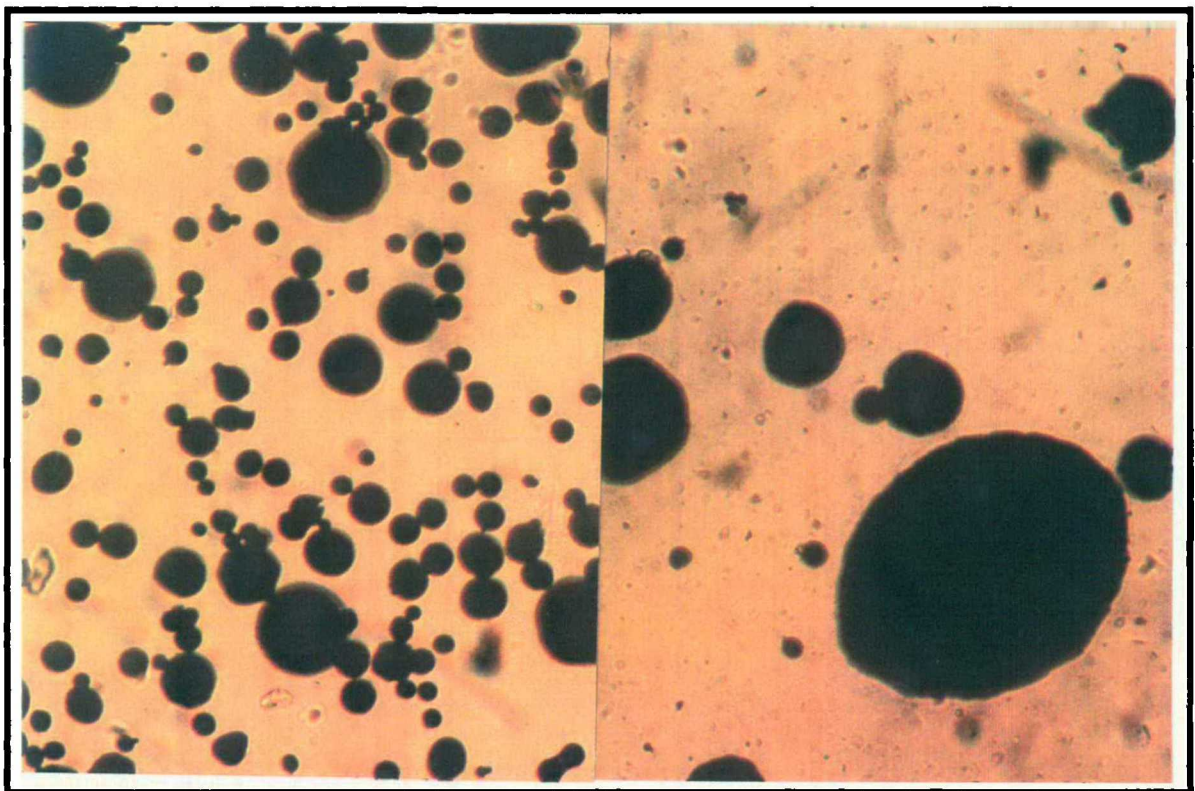


Figure 3.15. Photomicrograph of copper particles before and after overrolling (magnification x300)

Figure 3.13 clearly shows how some particles have increased in size during the overrolling process. A larger number of smaller particles are present in the final mixture; this could be the result of the generation of small fragments, or flattened disks measured at a sideways profile (the counter operates using a laser scattering technique). It should be noted that the apparatus may not overroll all the debris particles in the lubricant, some will pass around the contact region without being deformed.

3.3.3. Brittle Particle Damage

Six brittle debris materials were tested in the mangle rig. The nature of the surface damage on the contacting rollers is discussed in this section.

3.3.3.1. Indent Shape

The brittle materials left coarse irregular damage, with steep shoulders and rough surfaces. Figures 3.16 and 3.17 are 3D surface maps of the damage caused by glass and silicon carbide particles. Figures 3.18 and 3.19 show photographs of silicon carbide and Arizona dust damage.

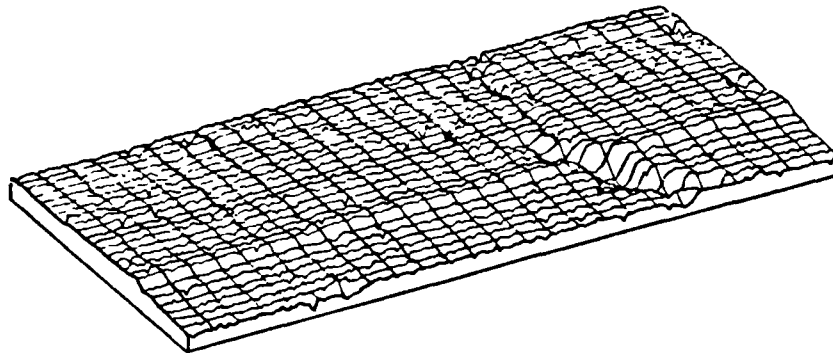


Figure 3.16. 3D surface map of roller surface damaged by glass ballotini (patch size 0.5mm x 1mm vertical magnification 25x horizontal).

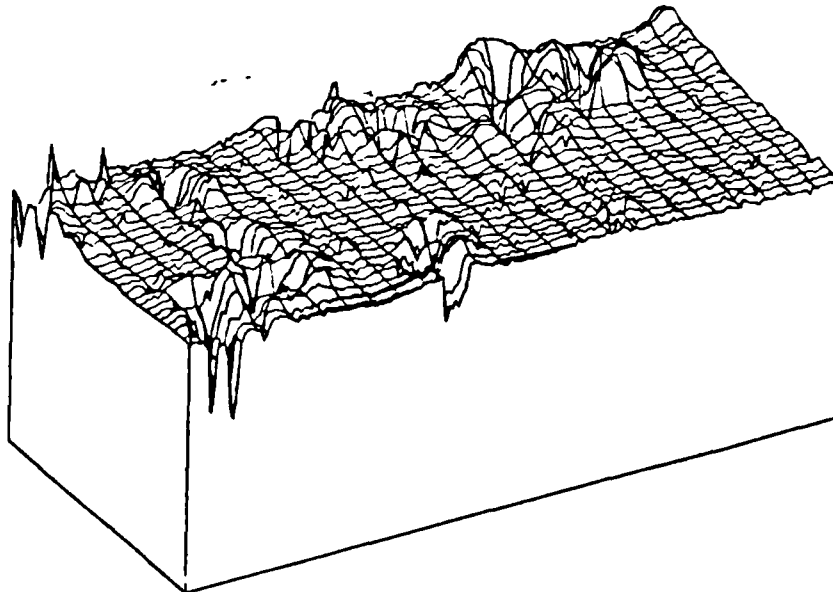


Figure 3.17. 3D surface map of roller surface damaged by silicon carbide (patch size 0.5mm x 1mm vertical magnification 25x horizontal).

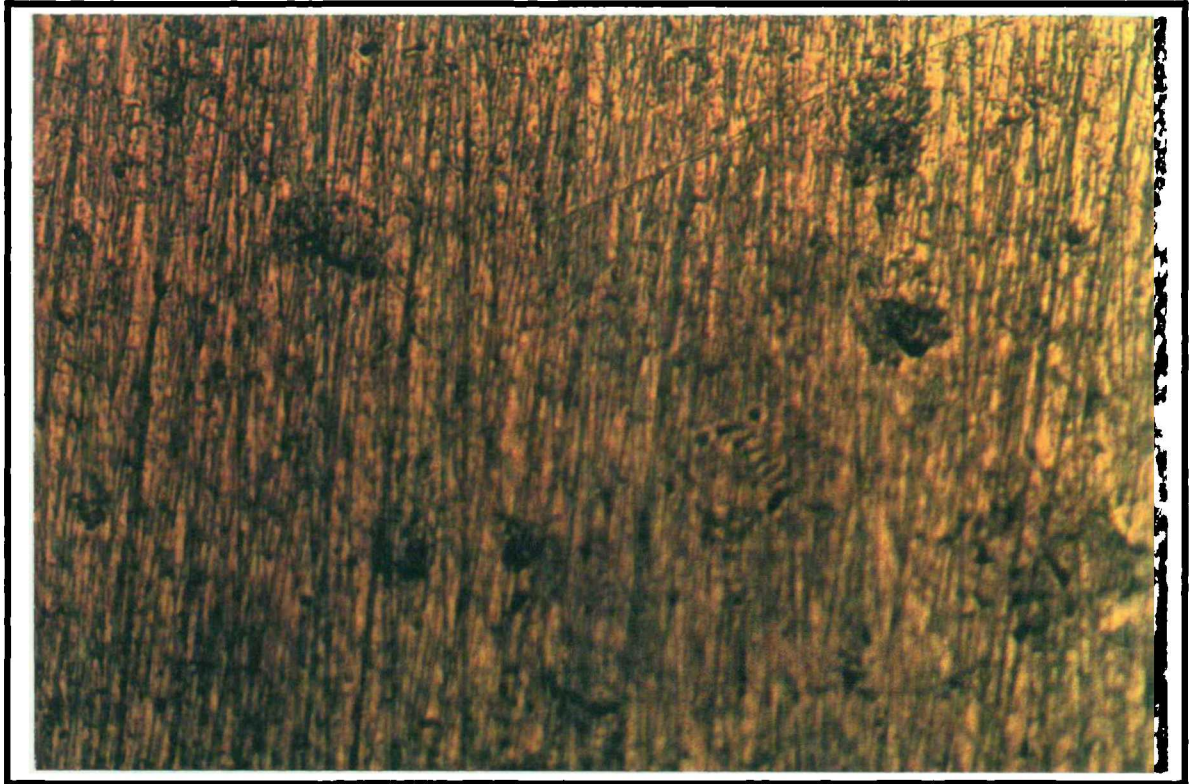


Figure 3.18. Photomicrograph of roller surface damaged by Arizona dust (magnification x300).

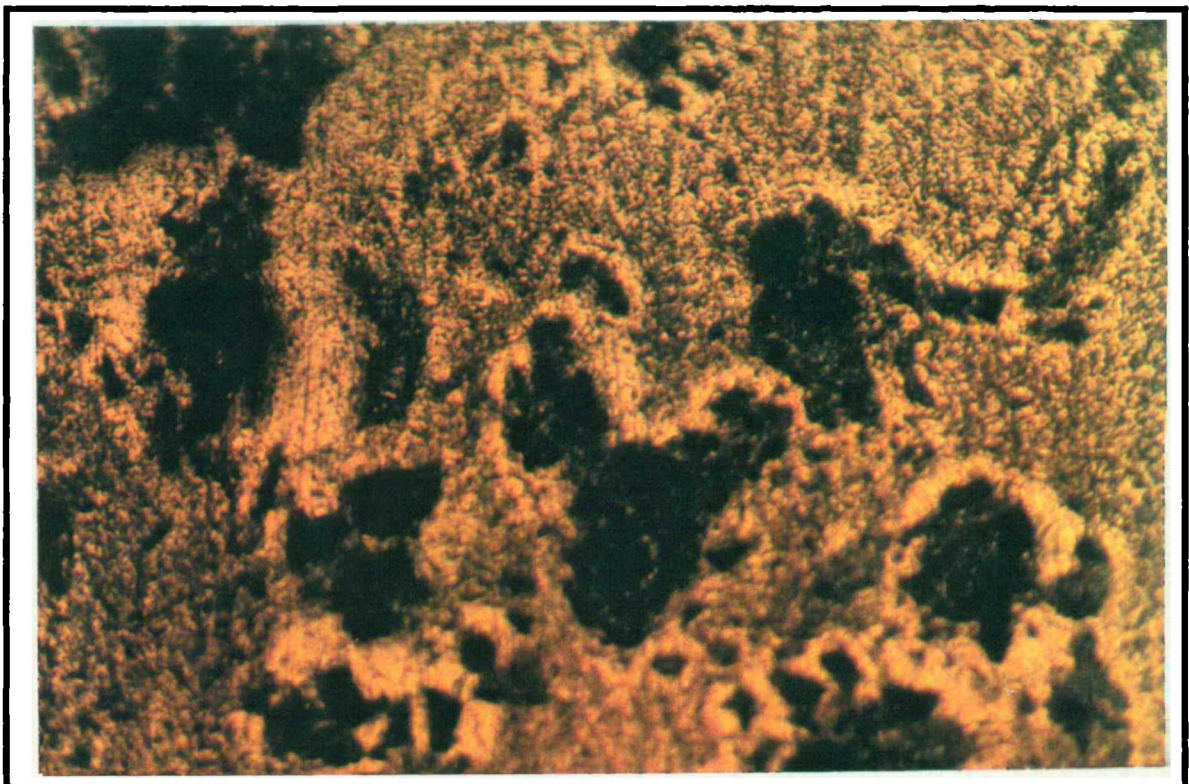


Figure 3.19. Photomicrograph of roller surface damaged by silicon carbide (magnification x300).

The low toughness brittle materials, Arizona dust and glass, left only a fine crazing on the roller surface. The damage was well dispersed and the contact surface fairly evenly covered with dents. In contrast, high toughness materials, like silicon carbide left very deep steep sloped dents; the damage sites were clustered together in patches.

These contrasting observations reflect the mechanism of particle breakdown in the entry zone (see §2.4). The brittle materials fracture readily and the small fragments disperse before indentation occurs. Thus, for brittle materials the size of the dents are largely independent of the original particle size, being dependent only on the fragment size. The tougher materials fracture further down the inlet region and the larger fragments tend to imbed as a mass causing extensive damage.

3.3.3.2. The Effect of Debris Fracture Toughness

The ease with which a brittle material breaks down is dependent on its fracture toughness. The degree of surface damage is also dependent on this property. Figure 3.20 shows the depth of surface dents compared with the approximate fracture toughness of the denting particles.

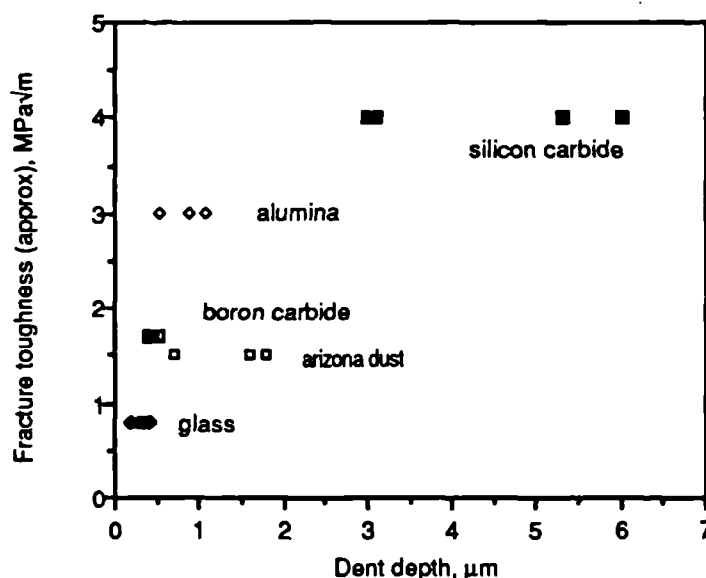


Figure 3.20. Dent sizes plotted against approximate fracture toughness of the debris materials (debris particle size 58-63μm)

The dent size clearly increases markedly with fracture toughness. The spread of data for dent sizes is likely to be attributable to the uncertainty of the materials properties, the statistical nature of brittle fracture, and the coarse particle size distribution.

3.3.3.3. The Distribution of Overrolled Particles

Again, the debris size distribution was measured before and after the test. Figure 3.21 shows the size distributions for alumina.

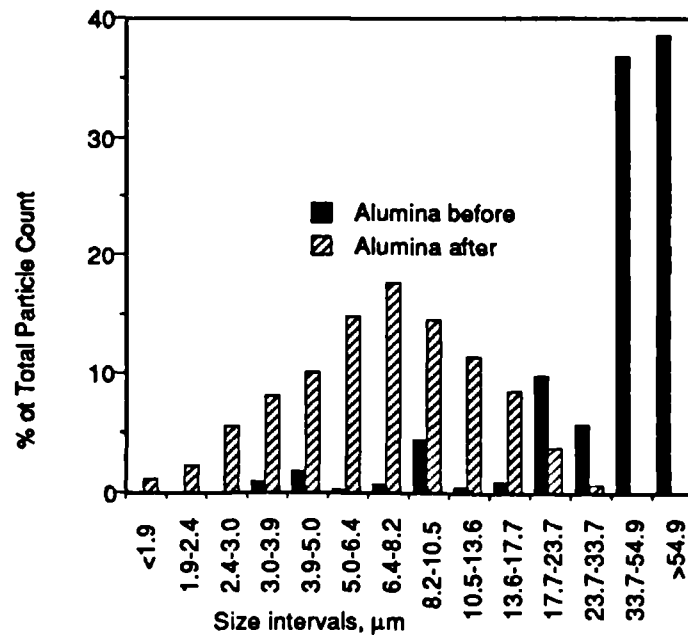


Figure 3.21. Particle size distributions for alumina particles before and after overrolling.

Clearly the large particles have been fractured into a population of smaller particles. Again, it should be noted that not all the particles may have been overrolled by the rolling element contact; some pass around the sides (although the regularity of the data suggests that the majority have been crushed). It is, therefore, not possible to use this data to determine an ultimate particle fragment size which will survive the contact pressures.

3.3.3.4. The Size of the Ultimate Fracture Fragment

It has already been stated that the indentation mechanism caused by brittle materials is critically dependent on the size of the fracture fragments which can survive the contact pressures. These become imbedded and leave surface dents. It is difficult to obtain a theoretical prediction of this final fragment size. The results of these experiments can help; examining the depth of the surface dents, and allowing for elastic recovery it is possible to estimate the size of the indenting fragment. Investigation of the graph included as Figure 3.20 suggest that values of $0.5\mu\text{m}$ for glass, $3\mu\text{m}$ for alumina, and $6\mu\text{m}$ for silicon carbide are not unreasonable.

3.3.3.5. The Effect of Particle Size

The mechanism presented for the overrolling and fracture of very brittle particles implies that the bulk of the surface damage is caused by the imbedding of the fracture fragments. Thus for a very brittle material like glass, the size of the dent should be independent of original particle size. To test this hypothesis experiments were performed using ballotini glass micro-spheres which were sieved into a variety of size bands and overrolled in the mangle rig. The size of the individual dents was measured using from surface profiles; to get an appreciation of the dent size several measurements must be taken and an average used (sample size 15). Table 3.4

shows the results. It thus appears that the size of the original particles makes little difference to the size of the final dent⁶.

Size Range, μm	Average Dent Depth, μm
<26	0.30
26-38	0.41
38-45	0.35
45-53	0.29
53-63	0.32
63-75	0.45

Table 3.4. Depth of average dents caused by ballotini glass micro-spheres of various size.

3.4. The Mechanisms of Debris Damage

The results of these experiments have been helpful in understanding how ductile and brittle debris particles can cause indentations when trapped between two surfaces. In this section the mechanisms implied earlier are explained in greater detail.

3.4.1. Ductile Debris Flattening

An explanation of how relatively soft ductile materials could indent hard surfaces was given by Hamer et al (1989). The process is represented schematically in Figure 3.22.

The first stage of the process is elastic, with Hertzian pressures building up between the surface and the particle. As the load increases first yield of the debris particle occurs. This point can be predicted from the elastic contact equations; and, for example, for a spherical particle occurs when the peak pressure, p_0 reaches about 1.6 times the particle yield stress, Y .

At this point, first yield of the particle occurs in a subsurface region. The particle continues deforming and the region of sub-surface plasticity extends. Eventually this region of plasticity extends to the surface of the particle and a state of, so-called, full plasticity is reached. The contact pressure is then equal to the particle material hardness.

Interfacial friction resists the lateral flow of the particle. The contact pressure builds up to overcome the frictional stresses. The pressure in the middle of the particle/surface interface is higher than at the edges (since the material in this region is more constrained).

As the load increases the peak of the pressure distribution increases. For a soft particle full closure may occur before counterface yield results (i.e. (a) in Figure 3.22). This represents the 'safe' case, where the particle overrolling causes no contacting surface damage. However, for a harder debris particle, extrusion continues and the pressure rises until yield of the counterface material occurs. As with the particle, counterface yield is initially sub-surface and then extends to the surface and full plasticity is achieved. The pressure distribution has then reached the hardness of the counterface material (as shown in (b) of Figure 3.22). The particle is deformed into a 'flying saucer' shape. The central region undergoes less plastic flow than the edges; in this region some of the geometrical closing of the surfaces is accommodated by plastic deformation of the counterface.

⁶ The extent of the damage is however very dependent on particle size; since a large particle will breakdown into many thousands of small particles (for example; a single $100\mu\text{m}$ particle fractures to give 10^6 $1\mu\text{m}$ particles!).

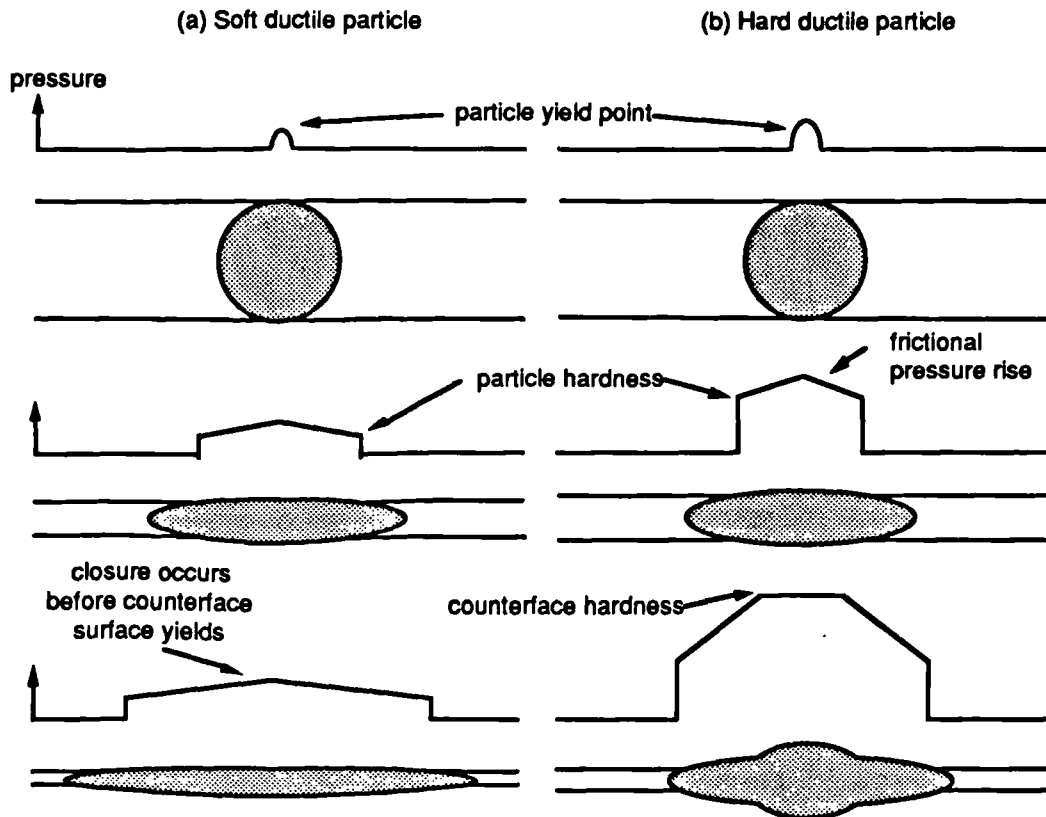


Figure 3.22. Schematic of the mechanism of indentation by (a) a soft ductile particle, and (b) a hard ductile particle. The diagram shows an idealised spherical particle with deformation and schematic contact pressures.

The loading continues with the region of plastic flow of the counterfaces growing until the surfaces conform around the flattened particle. The load is then supported by the debris material and the now contacting surfaces. Little further plastic flow of material occurs but the size of the elastic strains around the particle increase. This rearrangement of elastic strain reduces the amount of elastic recovery which can occur so deeper dents may result.

3.4.2. Brittle Debris Fracture

The first stage of the brittle debris overrolling process is the elastic contact between the surfaces and the particle. If the debris material is very brittle fracture will occur before yield of the counterfaces takes place (as in (a) of Figure 3.23).

As the counterfaces close further the fracture fragments are re-loaded; if they are above a critical size, subsequent fracture occurs. Eventually all the fragments will be fractured to below a critical size. The surfaces then plastically conform around them.

If the brittle particle is of sufficiently high toughness then yield, and possibly full plasticity of the counterfaces, will occur before particle fracture. Again the fragments will undergo successive fracture until the counterfaces conform around the final sized particles.

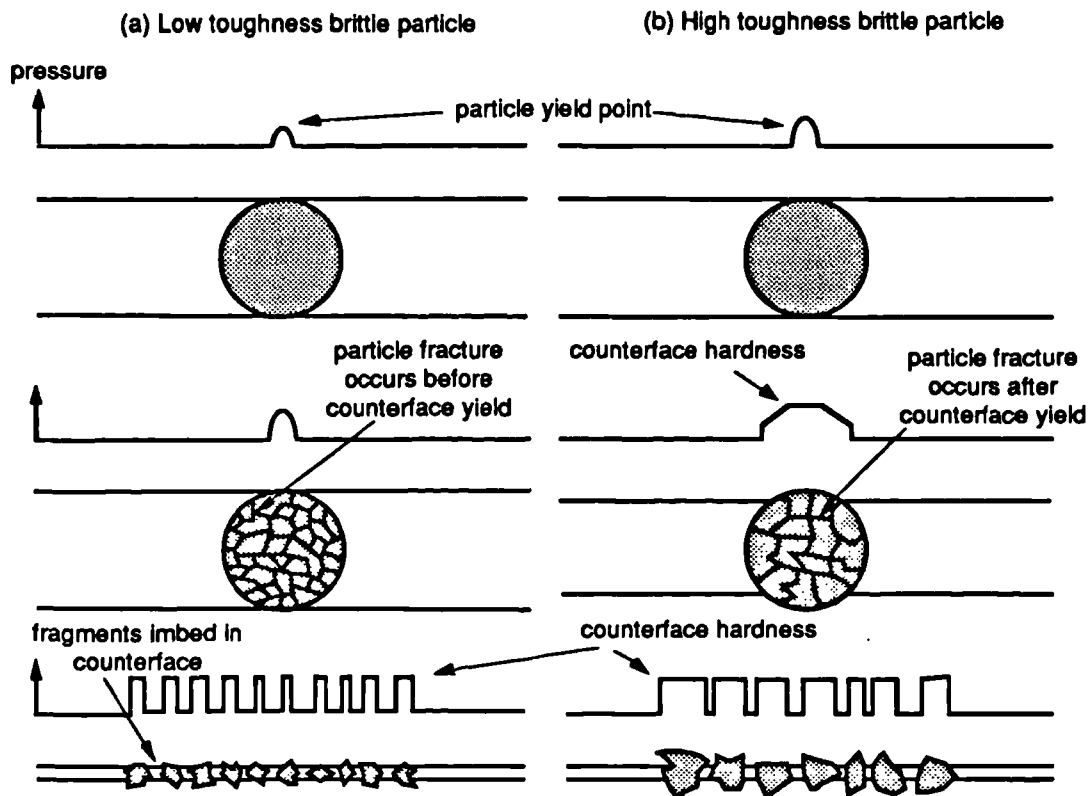


Figure 3.23. Schematic of the mechanism of indentation by (a) a low toughness brittle particle, and (b) a high toughness brittle particle. The diagram shows an idealised spherical particle fracturing and schematic contact pressures.

In the former process the only surface damage sustained is caused by the indentation of the final fracture fragments. The damage is therefore controlled by the size of the ultimate fragment only. In contrast the damage initiated by the crushing of tougher particles takes place at two stages; firstly, before the particle fractures, and secondly when the fragments become imbedded. In this case, the damage is controlled not only by the size of the ultimate fragment but also by that of the original particle.

3.5. Theoretical Modelling of Debris Damage

The modelling of the particle overrolling process is complex, involving elastic-plastic indentation, gross plastic flow, and fracture mechanics. However by making certain assumptions some predictions of debris damage can be made.

3.5.1. Ductile Debris Flattening

The work performed by Hamer, Sayles, & Ioannides (1987 & 1989) has extensively covered predictions of surface damage by squashing of ductile particles. No new material is added here; this section serves as a review of their work.

The first model considered the process as a fully plastic particle being extruded between elastic counterfaces. The particle was divided into concentric annular rings; the stresses on each elemental segment were resolved and a yield criterion introduced. An incremental pressure was derived; depending on whether the element was slipping (μP is less than the yield shear stress) or sticking (μP is greater than the yield shear stress, and then the particle shears just below the

interface). A relation for the elastic deformation of the counterface caused by the summation of annular rings of pressure was also determined. Iteration between these two relations gave the final equilibrium pressure distribution. This model was used to predict the onset of plastic deformation of the counterfaces for various particle hardnesses and sizes. This allowed maps detailing 'safe' and 'unsafe' debris particles to be generated. Figure 3.24 shows such a map.

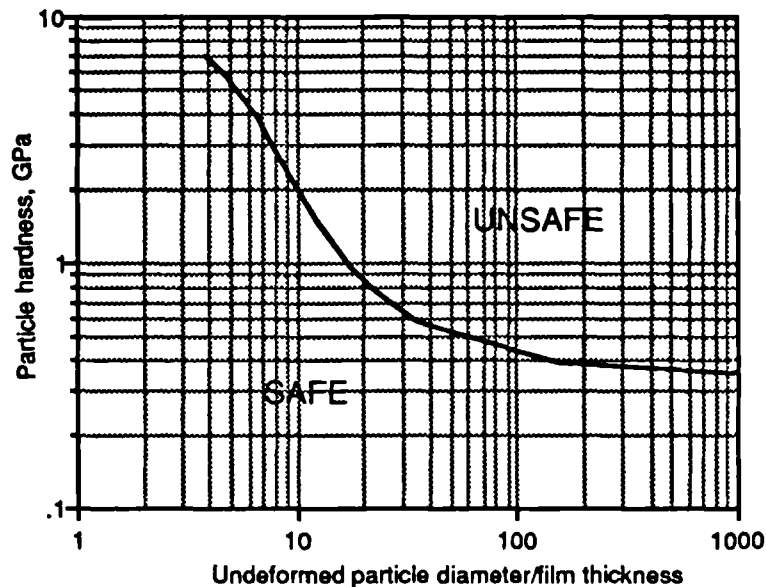


Figure 3.24. Map detailing safe and unsafe combinations of particle hardness and undeformed particle diameter to film thickness ratio from Hamer & Hutchinson (1992).

This model was then adapted ~~this model~~ to include plastic deformation of the counterface. The expression for the contact pressure acting over the extruding particle was replaced by one derived from the cavity model of Johnson (1970) and later from a finite element model (1992). This was then iterated with an extruding particle shape; when the pressure exceeded some yield pressure no further extrusion was assumed to take place in that region. In the next step only extrusion in the elastic counterface region was considered. The final deformed dent shape was determined by subtracting elastic deflections. The dent depths corresponded well with experiment (see §3.6) but the diameters were significantly less.

3.5.2. Brittle Debris Fracture

The brittle fracture of a debris particle is a complex process to model. However, it is possible, using some simple relations, to predict whether the particles fracture before surface yield, and what size the final fragments might be.

3.5.2.1. Particle Fracture Before Counterface Yield

The fracture of brittle materials occurs by the propagation of flaws within the debris particle. Tensile stresses cause the opening and therefore propagation of these cracks; whilst compressive stresses close them. Hence, brittle materials generally fail in tension. If the contact is elastic (i.e. no yield of the counterfaces has occurred) then the maximum tensile stresses

generally occur around the edge of the contact and in the plane of the surface⁷ For an idealised sphere on a flat half-space, Hertz (1882) theory (see for example Johnson 1985) gives this maximum tensile stress σ_t as;

$$\sigma_t = \frac{(1 - 2\nu) p_0}{3} \quad (3.1)$$

It is possible that fracture will initiate from any flaws in this region and the fracture stress, σ_f will be equal to this tensile stress. Of course fracture may initiate from some larger flaw elsewhere in the particle. The stress to cause fracture of a brittle material is largely determined by its fracture toughness, K_{IC} and the distribution of micro-cracks it contains (and particularly the size of the largest flaw or critical crack size, a_{cc}) by the following equation (Ashby & Jones 1986):

$$\sigma_f = \frac{K_{IC}}{\sqrt{\pi a_{cc}}} \quad (3.2)$$

The onset of counterface yield occurs when the peak pressure equals about 1.6 times the yield stress, Y (Johnson 1985). So combining equations (3.1) and (3.2) for fracture of the particle to occur before yield of the counterfaces;

$$a_{cc} < \frac{1}{\pi} \left(\frac{3K_{IC}}{(1 - 2\nu)1.6Y} \right)^2 \quad (3.3)$$

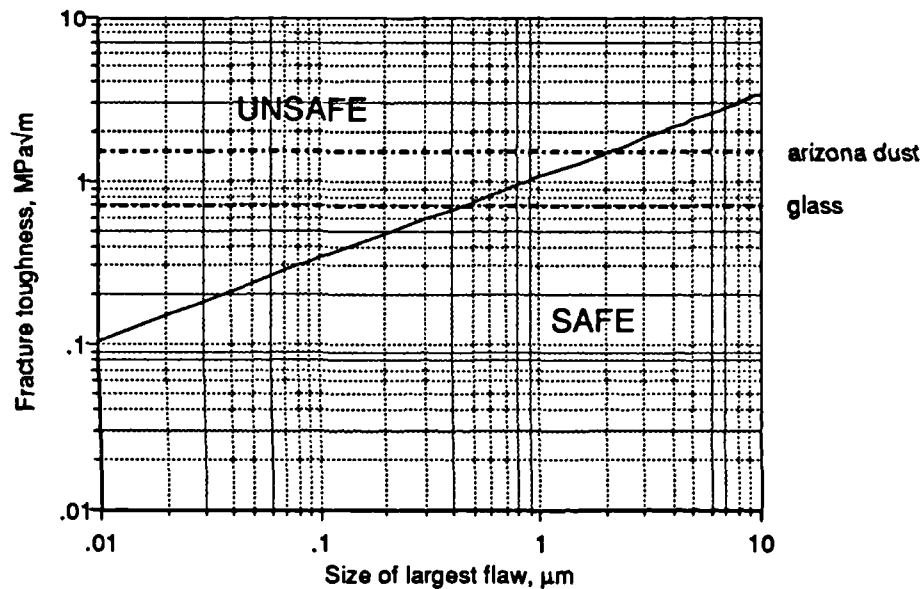


Figure 3.25. The fracture toughness of the debris material plotted against the size of the largest flaw to cause particle fracture before the counterface yield. As defined by equation (3.3).

⁷ It is these tensile stresses at the edge of the contact are responsible for the ring cracks which are produced when a steel ball is dropped onto a sheet of glass.

This relation is plotted as Figure 3.25 using the yield stress of bearing steel. The figure may be used as a map to predict whether a particle will cause surface yield before it fractures⁸; combinations of fracture toughness and debris flaw size below the curve are 'safe' whilst those above 'unsafe'. A low toughness debris particle with large surface flaws will fracture before the closing surface yield.

If we assume a flaw size of say $0.1\mu\text{m}^9$ for glass ballotini then yield of the counterfaces is likely to occur before the particle fractures. Indeed, this was the case in the experimental work. The photograph included as figure 3.26 shows smooth rounded dents caused by the overrolling of glass beads¹⁰. From the shape of the dent it is evident that counterface yield has occurred before particle fracture.

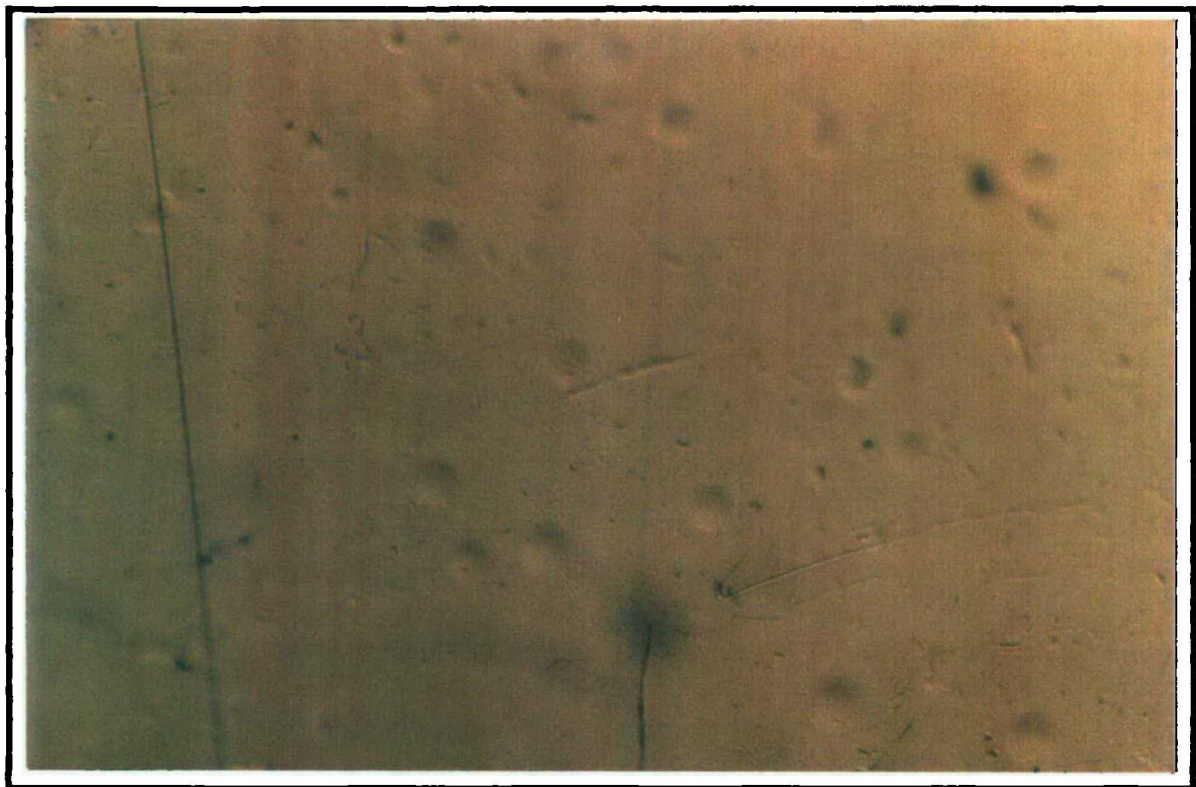


Figure 3.26. Photomicrograph of the rounded dents left by the overrolling of $60\mu\text{m}$ glass ballotini micro-spheres, indicating that counterface yield has occurred before particle fracture.

The pre-fracture dents in Figure 3.26 are on average $20\mu\text{m}$ in diameter. Since the circumferential elastic recovery of indentations is small (Tabor 1951), this measurement can be assumed to be close to the plastic indent diameter under load. Hence the contact area at which fracture of the glass particle occurred is defined. Figure 3.27 represents a single glass ballotini bead. The regimes represent the amount the sphere can be indented into the surface to cause; counterface yield, particle fracture, and counterface full plasticity (defined by $E^*a/YR=30$ according to Johnson 1985).

⁸ However, some surface yield will certainly occur after fracture, when the fragments become overrolled.

⁹ Surface profilometry suggests this is reasonable value, although of course a talysurf stylus may not penetrate fine surface flaws.

¹⁰ In addition to these rounded dents, on other regions of the specimen, exist the fine crazing left by the overrolling of glass fracture fragments.

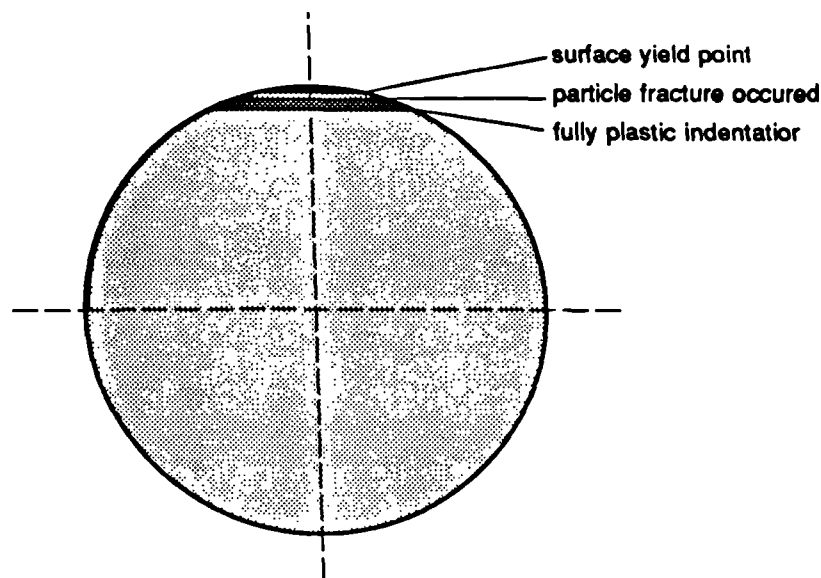


Figure 3.27. Schematic representation of a ballotini glass bead. The regions shown represent the indentation depths for counterface yield, particle fracture, and counterface full plasticity. Counterface yield occurs first followed by particle fracture. The full plasticity regime is not reached.

The analysis also demonstrates that it is unlikely that alumina or silicon carbide (with toughnesses of 3 and 4 MPa $\sqrt{m}$ respectively) will fracture before counterface yield occurs; particle flaws of about 8 to 10 μm would have to be present.

The calculation is valid for an idealised spherical particles; real debris materials are likely to be far from spherical. Jagged edges and peaks will lead to stress concentrations which are more likely to cause counterface yield.

The prediction of fracture strengths under conditions where counterface yield is exceeded is difficult. The finite element method could be employed to predict tensile stresses in particles being squashed in elastic-plastic contacts; this type of analysis is beyond the scope of this work.

3.5.2.2. The Ultimate Size of Debris Fracture Fragments

Another important variable is the size of the ultimate fracture fragments that remain after the particle is crushed. These will affect the size of the resulting damage, and for low toughness brittle materials, be solely responsible for surface damage. Defining this particle size is difficult because it depends on the microstructure of the particle and its relation to a complex stress field. However, estimates may be obtained by assuming that the particle cannot be reduced further than its critical crack size in the stress system. If we again assume an elliptical pressure distribution over a circular area but this time allow the peak stress to reach the hardness of the counterface material¹¹. The maximum tensile stress again occurs at the contact edge and is given by equation (3.1) replacing p_0 by the counterface hardness, H . Combining this with equation (3.2) gives;

¹¹ A more rigorous solution would be to use a uniform pressure distribution applied over the contact area (see Love 1929). This will give higher tensile stresses and therefore smaller fragment sizes; the present analysis is a simpler but more conservative approach.

$$a_{cc} = \frac{1}{\pi} \left(\frac{3K_{IC}}{(1-2\nu)H} \right)^2 \quad (3.4)$$

Figure 3.28 is a plot of this relation and Table 3.5 gives some calculated values of the ultimate fragment size.

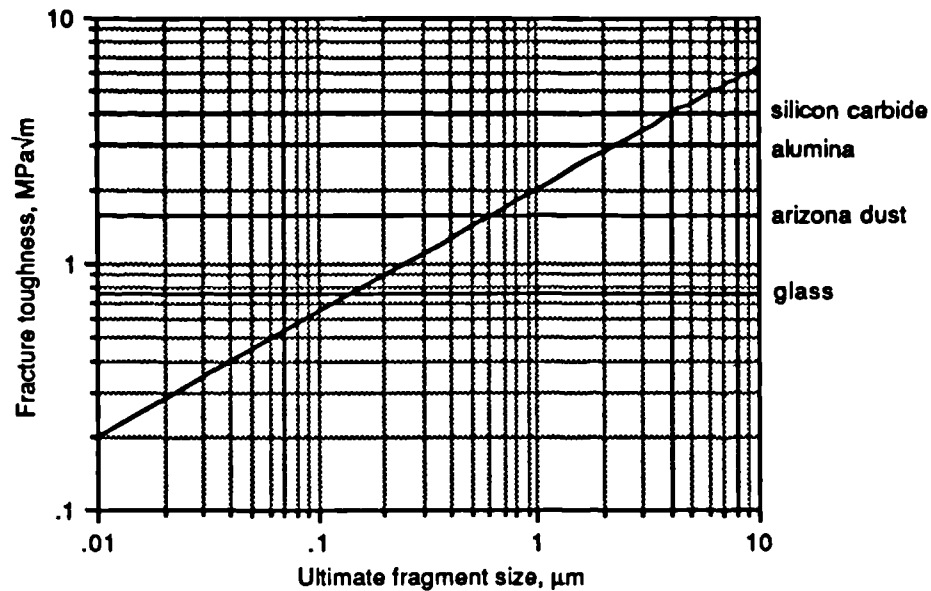


Figure 3.28. Prediction of the ultimate fracture fragment size from the fracture toughness of the debris particle material. Assumes that the debris particle is subjected to an elliptical pressure distribution with a peak value equal to the hardness of the counterface material.

Material	Fracture toughness MPa√m	Predicted ultimate fragment size μm	Experimental fragment size μm
Quartz	0.5	0.1	-
Glass	0.75	0.2	0.2-0.5
Arizona dust	1.5	0.7	0.2-0.9
Boron Carbide	1.7	0.8	0.2-0.5
Alumina	3	2.6	2-3
Silicon Carbide	4	4.7	5-10

Table 3.5. Estimates of the ultimate fragment size for various brittle materials, based on their fracture toughness and an assumed surrounding stress field. The predictions of equation (3.3) are compared with estimates based on experimental observations.

The values calculated are in general agreement with the estimates from the optical ehl studies (§2.4.2) and from the extent of the surface damage (§3.3.3.4).

3.6. Comparison of Debris Damage Results

In this section the experimental data and theoretical predictions are compared.

Table 3.6 shows the scaled anvil test data from Hamer et al (1989) (divided by 20 to scale from a 1mm pellet to a 50 μ m debris particle) compared with the results from the mangle rig and the particle forging analysis of Hamer et al (1992).

Debris material	Hardness GPa	Scaled anvil data		Mangle rig data		Hamer's analysis	
		depth, μ m	width, μ m	depth, μ m	width, μ m	depth, μ m	width, μ m
Aluminium	0.4	0.2	150	-	-	-	-
Copper	0.98	0.35	140	0.5-1.0	150-200	0.25	30
Mild steel	2.0	1.4	175	-	-	1.25	70
Stainless Steel	2.5	2.4	150	-	-	-	-
Low carbon steel	3.0	1.5	175	1.0-2.0	150-200	1.5	60
Med. carbon steel	4	-	-	-	-	2.25	90

Table 3.6. Comparison of anvil test data and mangle rig data with the predictions of the model of Hamer et al (1989).

The scaling of the anvil data corresponds reasonably well with the results of the mangle experiments. The anvil test is not a true analogue of the debris overrolling process; elliptical contact pressure, microslip, ehl, and rate dependent effects are not included. However, useful information can clearly be obtained. The forging model tends to under predict dent widths; Hamer attributes this to the zone of high elastic pressures surrounding the indentation during loading which allows yielding at the dent periphery leading to wider dents.

Table 3.7 shows a comparison of the scaled anvil data (from Dwyer-Joyce et al 1990) and mangle rig data for brittle debris materials. Also shown is an estimate of the ultimate fragment size.

Material	Scaled anvil dent depth, μ m	Scaled anvil width, μ m	Mangle dent depth, μ m	Mangle dent width, μ m	Ultimate fragment size, μ m
Silica	0.9	100	-	-	-
Arizona dust	-	-	0.6-1.0	50	0.2-0.9
Glass	-	-	0.2-0.3	30-40	0.2-0.5
Alumina	3.5	210	0.5-1.0	40-75	2-3

Table 3.7. Comparison of data from scaled anvil squashing tests with the mangle rig experiments.

In this case the scaled anvil data does not give a good prediction of the mangle rig dents. The fragments from a crushed pellet tended to indent the anvil surface as a mass; this has led to deeper dents than those found in the ehl case. The size scaling process is not valid for brittle materials; for this reason, and also because the strength of brittle materials depends on their size (which controls their flaw distribution).

Table 3.8 gives the data of Webster (1986) for debris dents in cylindrical roller bearings after the work of Bachu (1980). The debris was generated by circulating the test lubricant through a highly loaded gearbox; the particles were thought to be mostly metallic being removed from the gear wheel surfaces (of hardness 764V).

Material	Depth, μm	Width, μm	Aspect Ratio, μm
Gear box debris	1.0	130	130
Gear box debris	1.8	51	28
Gear box debris	5.8	364	63
Gear box debris	0.6	82	130
Gear box debris	3.0	123	41
Gear box debris	3.3	115	35
Gear box debris	3.6	102	28
Gear box debris	1.5	51	34
Gear box debris	1.3	25	19
Gear box debris	1.4	81	58
Gear box debris	4.8	174	36
Gear box debris	2.6	90	35
Gear box debris	4.4	212	48
Gear box debris	1.5	51	34

Table 3.8. Webster's (1986) data for debris initiated dents produced by the overrolling of gear box debris in a cylindrical roller bearing. (Approximate hardness of debris 764V).

This data for real gearbox debris compares favourably with the data from the ductile debris experiments. Dent depths and sizes are in reasonable agreement, given the unknown size of the particles responsible.

3.7. Conclusions

The purpose of this part of the study has been to define the relationship between the overrolled debris particle and the resulting surface damage. A purpose built rig was used to overroll realistic sized debris particles in an ehl contact. In both cases dent sizes were measured and related to the properties of the contaminating particle. The mechanism of the generation of surface damage is complex; some simple models are proposed to help predict the size of an indent given the nature of the debris particle.

The following conclusions have been drawn from this part of the study;

- (a) The scale of the surface damage caused by the overrolling of ductile debris is dependent on the size and hardness of the particle. The dents are rounded with a smooth surface.
- (b) The scale of the surface damage caused by the overrolling of brittle debris is dependent on the size and fracture toughness of the particle.
- (c) For low toughness ceramic materials, the particles may fracture before yield occurs. The size of the dents is then dependent solely on the size of the fracture fragments. The dents produced appear as a fine surface crazing of small steep sloped dents.
- (d) For tough ceramic particles yield of the elements occurs before fracture of the particle. The dents are deep with steep slopes and raised shoulders.
- (c) Scaled data from ductile pellet flattening tests (Hamer et al 1989) give good correlation with ductile debris dents formed in an ehl contact. This is not the case with brittle materials.

- (d) The theoretical models of Hamer provide a method for predicting the depth of ductile debris dents.
- (e) It is possible to estimate the ultimate size of fracture fragments produced when a brittle particle is overrolled in a concentrated contact. This result can be used to estimate the scale of surface damage caused by overrolling of very brittle debris materials.

CHAPTER 4

INDENTATION MODELLING

4.1. Introduction

The preceding sections of this study have described how debris particles behave in and around an elastohydrodynamic contact. Experiments have shown the nature of the surface damage generated during overrolling. This chapter describes some work carried out to model, in general terms, the indentation process. Finite element analysis has been used to generate indented surfaces and compare these to experimental results. Elastic, plastic, and residual stress fields have been generated. The next chapter describes how this information has been used in conjunction with a rolling contact fatigue life model to predict the reduction in life of rolling element bearings with indented surfaces.

The chapter begins with some background on plastic flow and the indentation problem. The finite element method is then discussed and the models described. The results are presented for two studies; a rigid indenter used to investigate elasticity, elastic-plastic transition, fully plastic behaviour, and residual stress state; and a spherical indenter model to generate indentations for comparison with experiment.

4.1.1. Plasticity and Plastic Flow

If a body is subjected to load its dimensions will change. The bonds between the molecules stretch or contract to allow extension or compression. The change in dimension is defined in terms of strain (extension divided by length) and the load applied defined in terms of stress (load divided by area). If the load is released the bonds revert back to their equilibrium position and the body returns to its original shape. This reversible process is called elastic deformation. If, however, the load is further increased, a point is reached where the deformation becomes permanent and the body no longer returns to its original shape. When this occurs, the material is said to have yielded and the deformation is termed plastic. Plastic flow is the result of the motion of dislocations in the crystal lattice allowing the large scale displacement of material.

In a simple tensile test experiment plastic flow occurs when the tensile stress exceeds the material yield stress. Under these stress conditions the deformation of the material is easy to predict (from a simple stress strain curve with or without strain hardening). In many engineering applications, such as indentation, the stresses are not uni-axial but combine in

various directions. It is necessary, therefore, to know when yield occurs under a combined stress field. Experimental observations show that yield does not occur under a hydrostatic stress state; it is only the deviatoric¹² stresses that are responsible for plastic flow. If a material is subjected to a combined stress state, where the principal stresses are σ_1 , σ_2 , and σ_3 , then it was found experimentally by Tresca (1864), that yield occurred when the maximum shear stress equalled a certain critical value, the yield stress, Y , so;

$$\max\{|\sigma_1 - \sigma_2|, |\sigma_2 - \sigma_3|, |\sigma_3 - \sigma_1|\} = Y \quad (4.1)$$

An alternative approach was developed by Von Mises (1913). He developed the strain energy criteria and suggested that yield occurred when;

$$J \equiv \frac{1}{6} \{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\} = k^2 = \frac{Y^2}{3} \quad (4.2)$$

Both criteria give similar results and are more applicable in certain practical situations (for example, annealed mild steel follows the Tresca criterion better); however, the Tresca criteria is slightly easier to apply.

Once the material has yielded plastic flow occurs and the displacements are no longer linearly dependent on the loading. Lévy (1871) and Von Mises (1913), working independently, suggested a relation between the strain increment, $d\epsilon$ and the orthogonal stress field which has become the basis of plasticity theory;

$$\frac{d\epsilon_x}{\sigma_x - \frac{1}{3}(\sigma_x + \sigma_y + \sigma_z)} = \frac{d\epsilon_y}{\sigma_y - \frac{1}{3}(\sigma_x + \sigma_y + \sigma_z)} = \frac{d\epsilon_z}{\sigma_z - \frac{1}{3}(\sigma_x + \sigma_y + \sigma_z)} \quad (4.3)$$

The equation expresses the isotropy of the body, the increment of flow in each direction is dependent on the deviatoric stress tensor in that direction. Thus to define the plastic flow during a loading cycle, it is necessary to determine the flow increments at all loading increments. Hence, the full loading history must be known; plastic flow is therefore a 'path dependent' process. The sum of the three strain increments is zero; plastic flow is therefore also a constant volume process.

The Lévy-Mises equation (4.3) uses the total strain increment, and is only applicable when elastic strains are zero. A later generalisation by Prandtl (1924) and Reuss (1930) allowed for both elastic and plastic strains. For a more comprehensive description of plasticity theory the reader is directed to Hill (1950) and Bridgman (1969).

4.1.2. Elastic Indentation and the Onset of Yield

Hertz (1882) developed the first analysis of the stresses and deformations produced by the contact of smooth elastic bodies. He assumed that the contact area would be elliptical and that the deformations would be small compared with the size of the bodies (hence the deflections may be determined by treating each body as a planar half space loaded over a small region). For the more general case of two solids of revolution in contact Hertz gives an elliptical distribution for the contact pressure, p :

¹² The hydrostatic stress component is defined as one third of the sum of the directional stress tensors. The deviatoric stress is the difference between the directional stress tensor and the hydrostatic stress at that point.

$$p = p_0 \sqrt{1 - \left(\frac{r}{a}\right)^2} \quad (4.4)$$

where p_0 is the maximum contact pressure, r is the radial position and a is the contact area radius. Applying this pressure profile to the two contacting bodies gives equations for the normal displacements. These displacements are related by the geometry of the two bodies and their distance of approach. The solution gives expressions for the contact radius; a ; and maximum pressure, p_0 :

$$a = \left(\frac{3PR}{4E^*} \right)^{\frac{1}{3}} \quad (4.5)$$

$$p_0 = \frac{3P}{2\pi a^2} = \left(\frac{6PE^{*2}}{\pi^3 R^2} \right)^{\frac{1}{3}} \quad (4.6)$$

where P is the normal load, R is the effective radius, and E^* is the reduced modulus. Expressions for the stresses, σ_r , σ_z , σ_θ within the solids may now be determined, see, for example Johnson (1985). The maximum stress difference occurs subsurface on the axis of symmetry at a depth of $0.48a$ and has a value $0.62p_0$ (for a Poisson's ratio $\nu=0.3$).

Applying the Tresca (or Von Mises) yield criterion, gives a peak contact pressure to cause the onset of yield;

$$p_{0Y} = 1.60Y \quad (4.7)$$

This first yield occurs at a point subsurface and is fully contained by elastic strains. Once yield has occurred the Hertzian solution is no longer valid and methods for modelling of plasticity must be employed.

4.1.3. Fully Plastic Indentation and Slip-Line Field Theory

The modelling of elastic-plastic flow is difficult; however, if gross plastic flow occurs then the elastic strains will be negligible compared to the plastic strains. Materials may then be represented as rigid perfectly plastic solids and plastic flow may be modelled using slip line field theory (developed by Hencky in 1923). Plastic flow is assumed to occur along orthogonal slip lines of constant shear stress.

Prandtl (1920) used this analysis for indentation by a planar rigid punch and predicted plastic contact pressures equal to 2.6 to 3 times the yield stress. Ishlinsky (1944) applied the theory to axi-symmetric indentation by a punch. He assumed that two of the principal stresses were equal and used a different yield criteria (the Haar-Karman criteria). These broad assumptions led to similar contact pressures.

4.1.4. Elastic-Plastic Indentation

When first yield occurs the plastic region is small and surrounded by material deforming elastically; the plastic strains are of the same order as the surrounding elastic strains. The

displaced material is accommodated by the elastic displacement of the surrounding material. As the load on the bodies increases the yield criterion is exceeded at more points and the region of plasticity expands. Eventually the plastic region extends to the surface and the displaced material is free to escape around the sides of the contacting bodies. This unconfined part of the process is characterised as fully plastic indentation.

The modelling of the elastic-plastic region of indentation is difficult since it requires the consideration of both elastic and plastic strains. Johnson (1970) in his 'cavity model' assumed that the indenter was encased in a hemi-spherical core of material under hydrostatic pressure. Outside the core the stresses and displacements have radial symmetry and are the same as those for a rigid-perfectly plastic body containing a spherical cavity under pressure (the solution for which was presented by Hill 1950). Using this analysis the elastic-plastic indentation pressures were correlated with indenter shape and expressions were developed for stresses throughout the body.

The most flexible analyses of elastic-plastic indentation have been developed using the finite element method. Hardy et al (1971) used this method to model the indentation of an elastic-plastic half-space by a rigid sphere and cylinder; they follow the growth of the plastic region and the contact pressures as the load increased. Computational difficulties arose when the fully plastic state was reached and plastic yielding became uncontained.

Follansbee and Sinclair (1984) achieved a model of a sphere indenting an elastic-plastic half-space well into the fully plastic region. The authors compared their results with experimental data and slip-line field models, good correlation was noted. The results demonstrate well the transition from elastic, through elastic-plastic, to fully plastic regions of deformation. At fully plastic loading the contact pressures flatten out to a uniform 2.8 times the yield stress. The paper includes plots of the growth of the plastic region, contours of octahedral strain, and stresses along the axis of symmetry and the surface. Johnson (1985) provides a useful summary of this data in comparison with other FE solutions, analytical solutions, and experimental data.

Residual stresses were also presented; hoop and radial stresses were found to be tensile, to a value of 0.35 times the flow stress, within the contact area. These tensile stresses were limited to the surface regions only (reaching 0.10a to 0.13a beneath the surface). Plots of the surface displacements show 'sinking-in' during loading and 'piling-up' when full plasticity is reached. Shallowing of the indent by about 6% was noted on unloading; the diameter of the indent showed little change.

Johnson (1968) measured plastic strains by scribing fine lines on the surfaces of spherical copper test pieces. He deduced the contact pressure distribution (and hence one of the principle stresses) from the deformed shape of the surfaces. Knowing this and the surface strains he was able to produce surface distributions of plastic radial and hoop stress. The superposition of an elastic unloading gave the surface residual stress profiles.

Olver et al (1986) used X-ray diffraction to measure residual stresses following the plastic contact of two spheres. The results for surface residual stresses were good and showed a close resemblance to the theoretical predictions (despite the low resolution of the system). The results of this analysis and that of the previous author are presented alongside the present finite element studies in §4.3.4.

4.2. The Finite Element Method and Plastic Contact Problems

It was decided that finite element analysis would be the most flexible method for obtaining data on the debris indentation problem. Because of the nature of the problem (high stress gradients, changing contact conditions, and plastic flow) it is necessary to check the results of such an analysis very carefully; it is easy to introduce numerical errors which magnify and upset the results. It is instructive therefore to understand some of the basic principles of the finite element method and the special requirements for modelling contact and plastic flow.

4.2.1. The Finite Element Method

The finite element method divides a complex loaded solid body into a number of elements (triangular, quadrilateral, tetrahedral or bricks) joined to each other at nodal points. The displacement at each nodal point is chosen as the unknown variable. A shape function is chosen to describe how the displacements vary over each element (i.e. linearly or quadratically). Strains are obtained from strain-displacement relationships. Stresses are obtained from the strains by the constitutive equations. For linear problems;

$$\{F\} = [K]\{\delta\} \quad (4.8)$$

where $[K]$ is referred to as the stiffness matrix which relates the displacement of the nodes $\{\delta\}$ to the matrix of external forces, $\{F\}$. Boundary conditions are applied and the system of linear equations is solved (usually using Gaussian elimination or Gauss-Seidel iteration). Once the displacements at each node point have been determined the stresses and strains may be calculated.

When the displacements of the nodal points are proportional to the applied load the problem is linear. Many engineering problems are non-linear; the plastic indentation problem is such a non-linear problem. The contact between two surfaces is non-linear; when the contact is closed normal (and shear stresses) may be transmitted, re-opening implies no contact pressure; the stress state is dependent on the contact and therefore load history. Finite element programs use gap or interface elements to model the contact between two bodies. At each iteration the contact condition is inspected along with the usual checks on equilibrium.

Coulomb friction at the interface may also be modelled; if the contact is sticking shear stresses are transmitted. When the shear stress exceeds a limiting value (given by μP) the contact slips. This stick-slip process is also dependent on the loading cycle; once a pair of contacting points has undergone relative motion the process cannot be reversed.

Plastic flow is also path dependent; the magnitude of plastic strain is dependent on the loading cycle (loading from A to B to C and back to B does not generate the same plastic flow as loading from A to B). It is therefore necessary to follow the entire loading process to give the correct stresses and strains at the end. Finite element modelling of plastic flow is possible. A yield function defines the limit of elastic response and is a function of the stress and work-hardening state. The post-yield strains are divided into an elastic and a plastic part. The elastic part responds linearly whilst the plastic part is defined by the Prandtl-Reuss flow rule (as described in §4.1.1). Further details can be found in the ABAQUS theory manual (1989).

In the analysis of the indentation problem we have three sources of non-linearity; contact, friction, and plasticity. The final stress state is dependent on the loading history. A rigorous numerical analysis of these path dependencies would require many discrete load steps. In

practice this process is costly in terms of computer time, and the usual method is to run the analysis with progressively smaller load steps until the final solution remains unchanged.

4.2.2. ABAQUS

The general purpose finite element code ABAQUS was chosen for the analysis in this study¹³. The code was developed in 1978 by Hibbitt, Karlsson, and Sorensen Inc. It was designed with a particular emphasis on complex non-linear problems and has a good reputation amongst FE users. The code, written in FORTRAN, is arranged into modules to solve various classes of problem. Modules exist for static and dynamic analysis of 2D and 3D problems incorporating; contact, plastic flow, heat transfer, creep and rate dependency, combined heat transfer and stress analysis, acoustics, modal analysis, or J-integral evaluation (see Brebbia 1985).

The finite element problem is specified in an 'input deck' consisting of a number of keywords. A typical deck for a contact problem is divided into;

- (a) Model definition cards define the nodal co-ordinates and element connectivity. Nodes and elements are divided into sets for convenience.
- (b) Material properties cards defining elastic and plastic properties attributable to the element sets.
- (c) Contact specification cards give the friction conditions and associated properties.
- (d) Boundary condition cards fix nodes and node sets in any required degree of freedom.
- (e) Step definition cards define the analysis to be performed and parameters for the control of the iteration process.
- (f) History definition cards give the loading sequence in terms of prescribed displacements, concentrated loads, or distributed loads.
- (g) Output control cards specify the format for the output of data to give plots, tables of data, or input files for a pre-processing package.

Mesh generation and the post-processing of results was performed using FEMGEN and FEMVIEW developed at Leicester University. The program provides a simple user interface for mesh creation and display of results. The FEMVIEW/ABAQUS interface program compiles an input file which is then run with ABAQUS.

The programs were run on a DEC2100 under ULTRIX 3.0 (DEC's implementation of the UNIX operating system).

4.2.3. Finite Element Modelling of Surface Damage

This study is concerned with the entrapment, overrolling, and deformation of debris in a concentrated contact. Ductile debris flattens out into platelets (see §3.4.1). The finite element modelling of this type of gross plastic flow is difficult. The mesh distorts rapidly into long thin elements which are not suitable for the interpolation algorithms; continual remeshing is

¹³ The finite element code PAFEC and the pre-processors PIGS and SUPERTAB were also evaluated but found to be less flexible for the required type of plastic contact problem.

required. This process is time consuming and costly. The ductile forging model adapted by Hamer (1989) described in §3.5.1 is a more useful approach.

The work presented here is thus confined to rigid indentation or indentation by hard particles. This more closely models the overrolling of brittle debris particles where the fragments indent rigidly into the bearing surfaces (see §3.4.2).

The data from the finite element indentation studies was then used in the prediction of fatigue life. The residual stresses from a model indentation were used in conjunction with a rolling contact fatigue model to predict the reduction in life of an indented surface.

Finite element analysis is a dangerous tool. It is all too easy to generate meaningless solutions from simple specifications. A badly designed mesh or a poorly modelled boundary conditions can give wildly inaccurate results; particularly with problems with large non-linearities where errors can be magnified at each iteration. Therefore, the first studies carried out, were to verify the results of some indentation analysis against other existing experimental and theoretical data.

4.3. The Spherical Indentor

The first meshes created were axi-symmetric sphere on flat half space analyses. The finite element modelling of elastic, elastic-plastic, and plastic regimes of indentation were investigated. Of particular interest were the displaced shapes and the residual stresses; which are important in the predicting of rolling contact fatigue life.

4.3.1. The Elastic Regime

A mesh was generated to represent the axi-symmetric contact of a ball on a flat (see Figure 4.1). The two bodies were given the materials properties of a typical bearing steel (Young's modulus of 209GPa, Poisson's ratio 0.3).

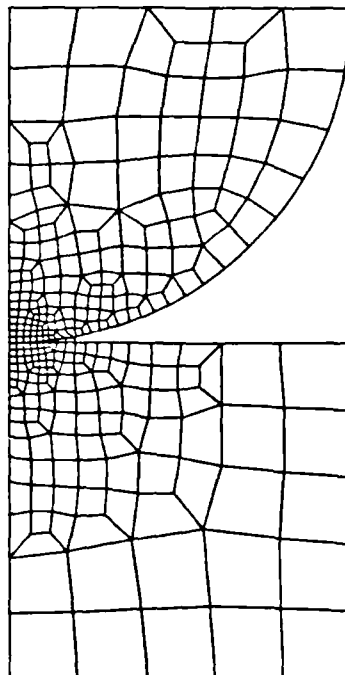


Figure 4.1. Typical finite element mesh used for elastic spherical indentation modelling.

The base of the lower body was simply supported whilst the sides of the body were constrained to only vertical motion¹⁴. The nodes at the top edge of the upper body were constrained to equal vertical motion (i.e. all these nodes move down in a horizontal line). Loads were applied to the node on the axis of symmetry (see Figure 4.2).

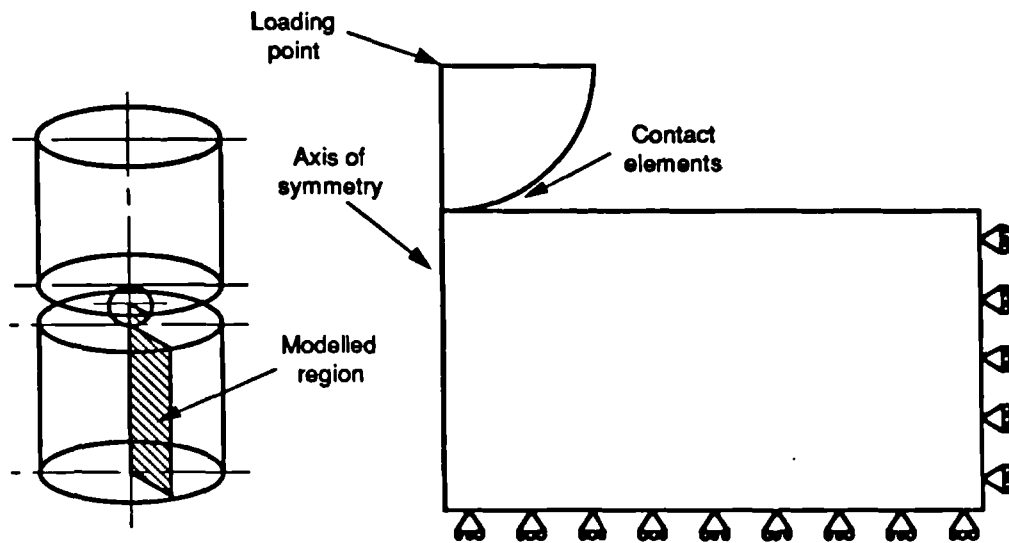


Figure 4.2. Boundary conditions and loads applied to the finite element mesh.

Eight noded axis-symmetric quadrilateral reduced integration elements (designated CAX8R) were used; the contact was modelled using six noded interface elements (designated INTER3A). The eight noded elements describe a parabolic shape function and are therefore better for modelling the high stress gradients of a Hertzian contact. In the later analyses, some convergence problems occurred and were solved by using four noded quadrilateral elements in the mesh.

The first models used increasingly refined meshes to compare the results. The first mesh contained 368 elements, the second 522, and the third 614 elements. Figure 4.3 shows a plot of the contact pressures generated by loading the two bodies together compared with the Hertzian elastic solution (the data is normalised using the Hertzian peak pressure and contact half-width).

The coarser meshes tend to over predict the peak contact pressure and overshoot as the bodies come out of contact. The reason for this behaviour comes from the formulation of the FE solution. The solution is determined at the integration (or Gauss points) which are located within an element. The nodal stresses are then determined by extrapolating from these points to the corners of the element; a high stress gradient causes over or under prediction. It is necessary, therefore, to have sufficient elements such that the stress profile may be accurately represented by each element's linear fit.

¹⁴ The size of the half space is large enough such that the boundary conditions at the edges are irrelevant; slides or free movement could have been applied. For very large indentations these boundary conditions may affect the results; free movement leading to a less constrained system and therefore lower contact pressures and slides leading to a more constrained system and higher pressures.

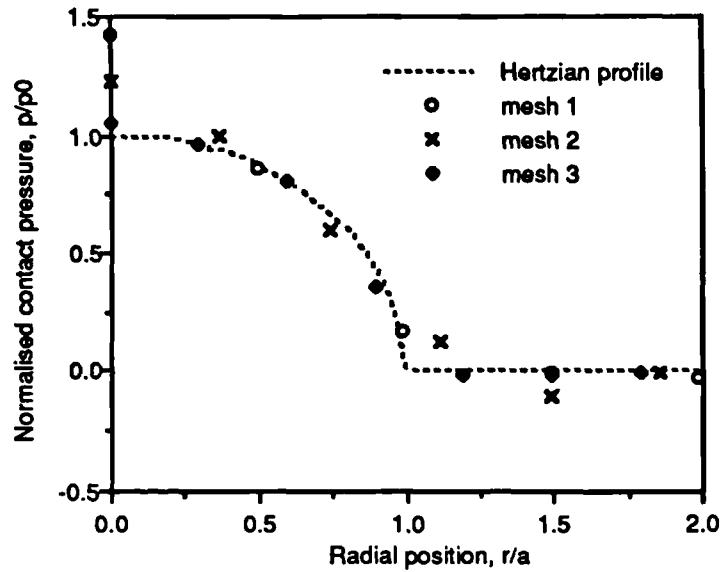


Figure 4.3. Contact pressures generated from three finite element models compared to the equivalent Hertzian solution.

Figure 4.4 shows a plot of the Mises, radial, hoop, shear, and axial stresses along the axis of symmetry (these are principal stresses). The Hertzian solutions for these stresses are given by, for example Johnson (1985);

$$\frac{\sigma_r}{p_0} = \frac{\sigma_\theta}{p_0} = -(1 + \nu) \left\{ 1 - \left(\frac{z}{a} \right) \tan^{-1} \left(\frac{a}{z} \right) \right\} + \frac{1}{2} \left(1 + \frac{z^2}{a^2} \right)^{-1} \quad (4.9)$$

$$\frac{\sigma_z}{p_0} = - \left(1 + \frac{z^2}{a^2} \right)^{-1} \quad (4.10)$$

$$\tau_1 = \frac{1}{2} |\sigma_z - \sigma_r| \quad (4.11)$$

These relations are plotted alongside nodal data from the finite element solutions on Figure 4.4.

The peak principal shear stress is correctly predicted as $0.31p_0$ at a position $0.48a$ below the surface. The onset of yield occurs when the peak contact pressure equals $1.6Y$; at this point the Von Mises stress at the subsurface location will equal Y . Scaling the finite element prediction shows that first yield will be predicted to within an accuracy greater than 99.5%.

Provided the mesh is sufficiently refined the accuracy of the elastic solutions are good. Stresses, contact areas, and hence the onset of yield are accurately predicted. This elastic solution will then be used as a starting point for the plastic calculations.

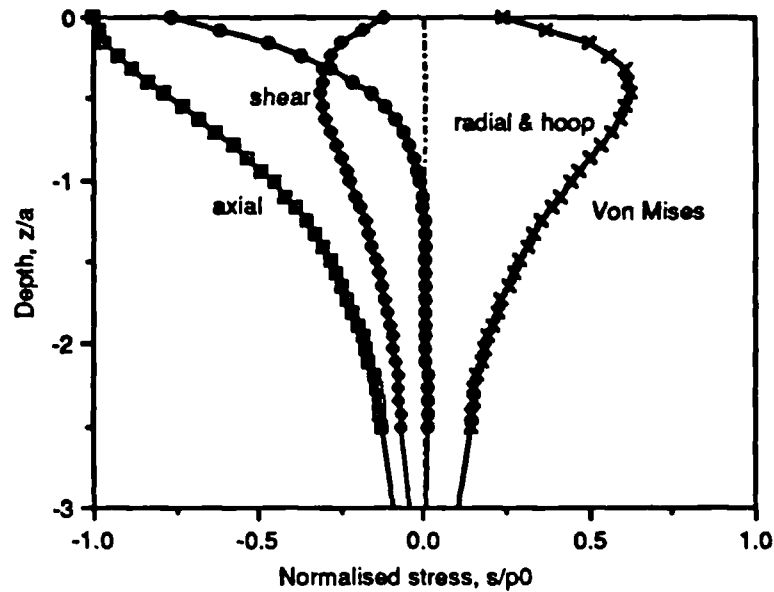


Figure 4.4. Elastic stress distributions along the axis of symmetry from the finest finite element mesh compared to the Hertzian solution (lines represent the Hertzian solutions, data points the FE solution).

4.3.2. The Elastic-Plastic Transition Regime

A rigid indenter mesh was used to study the transition from elastic to fully plastic indentation. Figure 4.5 shows the mesh; eight noded axi-symmetric quadrilateral reduced integration elements (designated CAX8R) were used.

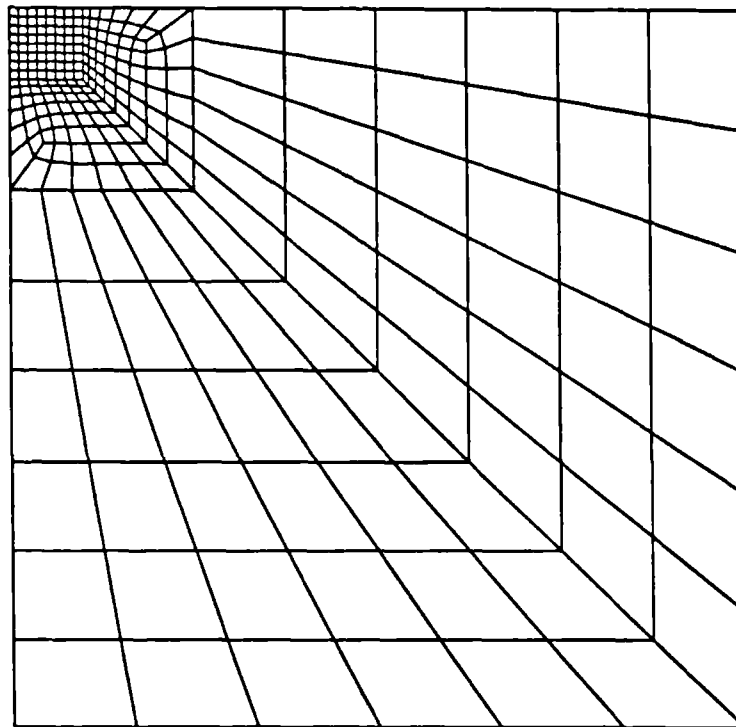


Figure 4.5. Rigid indenter finite element mesh used for the study of elastic-plastic indentation

An axi-symmetric rigid surface was defined and the contact modelled using rigid surface interface elements (designated IRS3A). Again the body was given the properties of a typical bearing steel (this time with a yield stress of 2.8GPa but no strain hardening). The base of the lower body was simply supported and the indentation processed by displacing the rigid surface onto the lower body.

The loading was divided into eight steps, the first step being elastic at the onset of yield (the first step of any plasticity analysis should always be fully elastic to give a good foundation for the later plastic steps). The last step is an unloading step and the results of this part of the analysis are presented in §4.3.4. ABAQUS further sub-divides the loading into smaller increments; typically an indentation from elastic to fully plastic may take between 25 and 100 increments depending on the complexity of the mesh.

As the indentation progresses the subsurface yield region grows; until it reaches the free surface and the process is termed fully plastic. Figure 4.6 shows a plot of the equivalent plastic strain (this is defined as the total accumulated plastic strain and can be used to define the yield surface) down the axis of symmetry for each of the seven loading steps.

A useful parameter in the elastic-plastic spherical indentation process is E^*a/YR , where a is the radius of the indentation. Johnson (1970) showed that the indentation hardness is governed solely by this non-dimensional strain parameter. Indentation experiments and models can therefore be correlated using this parameter (as in Figure 4.8).

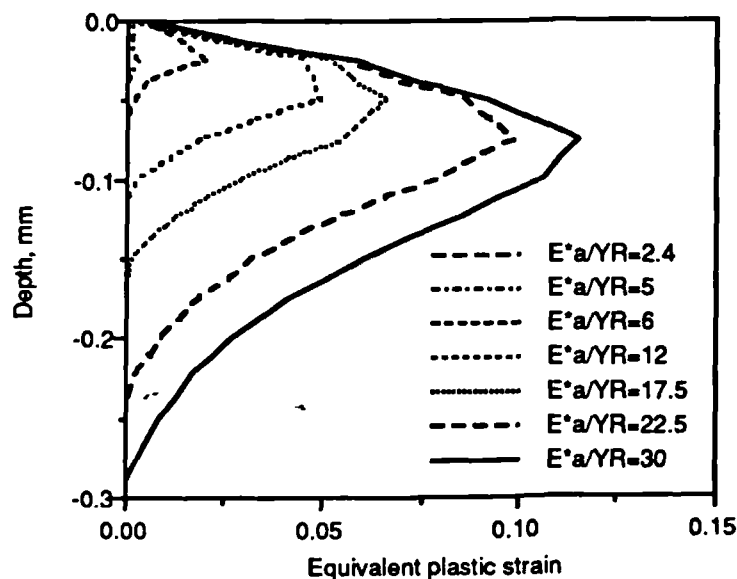


Figure 4.6. Equivalent plastic strain distribution along the axis of symmetry for each of the loading steps. The equivalent plastic strain shows the extent of yielding.

The growth of the plastic zone during the indentation can be seen; the zone breaks through to the surface at the sixth load step (when $E^*a/YR=22.5$).

Figure 4.7 shows the contact pressure profiles generated at each of these loading steps.

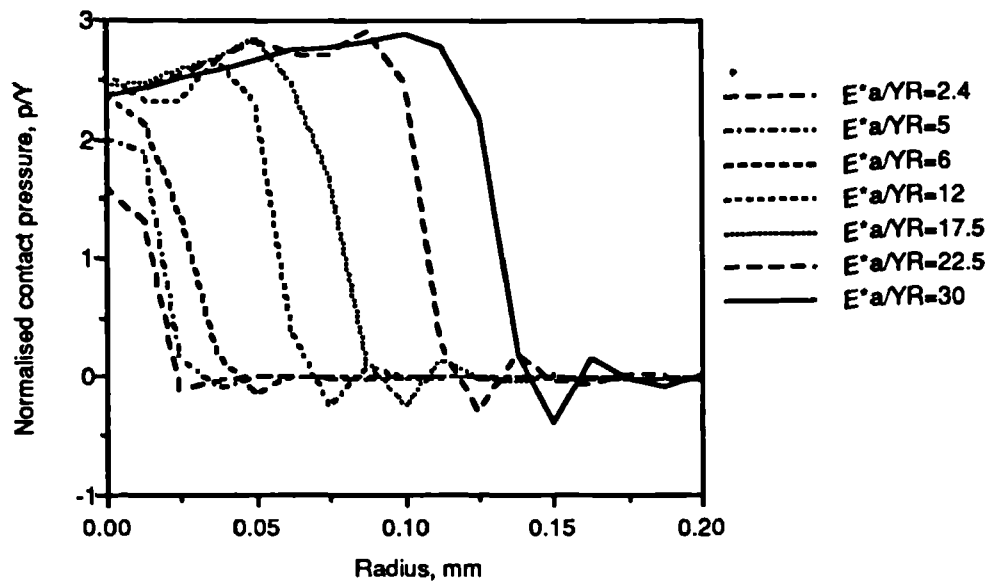


Figure 4.7. Development of elastic-plastic contact pressures during the indentation.

The first load step is the limit of the elastic case and the Hertzian parabola has a maximum at $1.6Y$. This develops into an almost uniform pressure distribution at about the sixth load step; this pressure has a value of about $2.6Y$ (the perfectly plastic slip line field analyses, as described in §4.1.3, predict about $2.8Y$). The pressure tends to fall off slightly towards the centre of the indentation. Outside the loaded region the pressure should fall to zero; the finite element solution overshoots slightly and oscillates back to zero. This is again a function of the interpolation method which over predicts in regions of high stress gradient. A finer mesh in this region would reduce this effect.

Figure 4.8 shows a plot of this mean contact pressure against the non-dimensional strain variable E^*a/YR . Use of this parameter allows the comparison of indentations of different size on various materials; Johnson (1985) plots data in this manner. The finite element results from this analysis are shown as diamonds. The results from later experimental and FE studies of tungsten carbide (WC) indentors, described in §4.4, are plotted as crosses and squares. Tabor's (1951) indentation experiments are also plotted. The lines on the graph show the elastic, cavity model, and slip line field solutions.

It proved difficult to achieve larger indentations in the finite element model presented here. The high yield stress of the material used required large strains and hence extensive mesh deformation to achieve full plasticity. The mesh became too deformed to take the analysis further into the fully plastic region (i.e. to higher values of E^*a/YR). The analysis was therefore repeated using a material of lower yield stress (modelled as mild steel with $\sigma_y=300\text{MPa}$). This softer model was deformed further into the plastic region; this soft indenter data is marked on the graph as triangles.

The agreement between the finite element and theoretical results is good. The onset of yield is predicted accurately. According to Johnson (1985), the onset of full plasticity occurs when $E^*a/YR \sim 30$. The FE solutions achieved full plasticity slightly earlier (at $E^*a/YR = 22.5$). The deviation in the tungsten carbide indentation experiments is possibly due to an uncertainty in the materials properties.

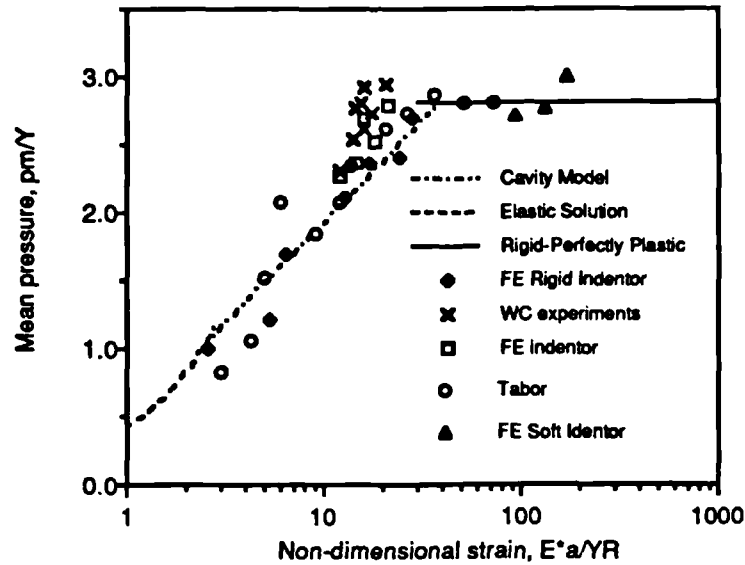


Figure 4.8. Graph of the mean pressure of indentation against non-dimensional strain parameter. Finite element results are compared with Tabor's experimental data and elastic, cavity model and slip line field solutions.

An alternative approach for comparing indentation data is presented in Figure 4.9. The plot is the load normalised by the load to cause first yield against the displacement normalised by the displacement at first yield (i.e. a compliance curve).

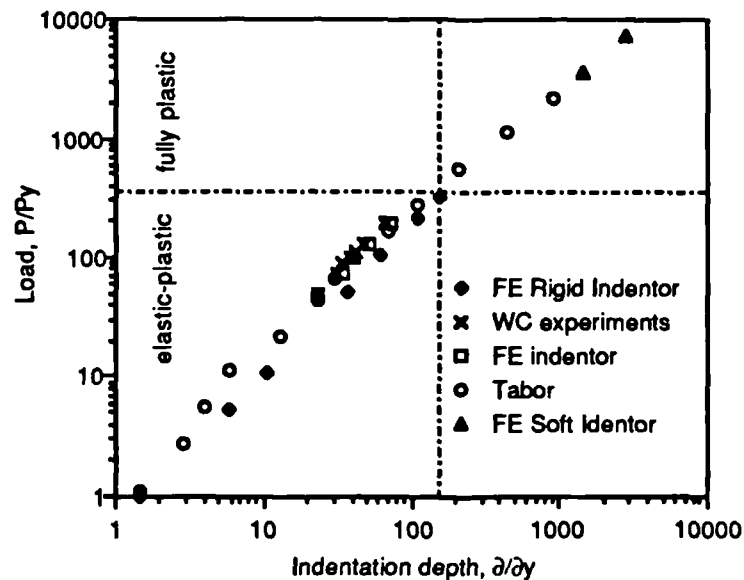


Figure 4.9. Graph of the load normalised by load to cause first yield against displacement normalised by displacement at first yield. Finite element results are compared with Tabor's experimental data and tungsten carbide indentations.

In the elastic regime $P \sim \delta^{2/3}$; in the fully plastic regime the data lie on an approximately straight line, Johnson (1985) suggests that in this regime $P/P_y \sim 5.5 \delta/\delta_y$.

The graphs included as Figures 4.10 and 4.11 show the radial and hoop stress profiles at the half-space surface for each of the loading steps. The profiles develop into uniform distributions in a similar manner to the contact pressure profiles. Tensile hoop stresses are generated at the edge of the contact region whilst radial stresses are compressive; this is the opposite case to the elastic stress state. This change in stress state from tension first radially and then circumferentially is responsible for the change in fracture mode for brittle and semi-brittle materials; glass fails at low stress with ring cracks caused by the elastic tensile radial stresses; whilst perspex fails with radial cracks caused by the higher plastic tensile hoop stresses.

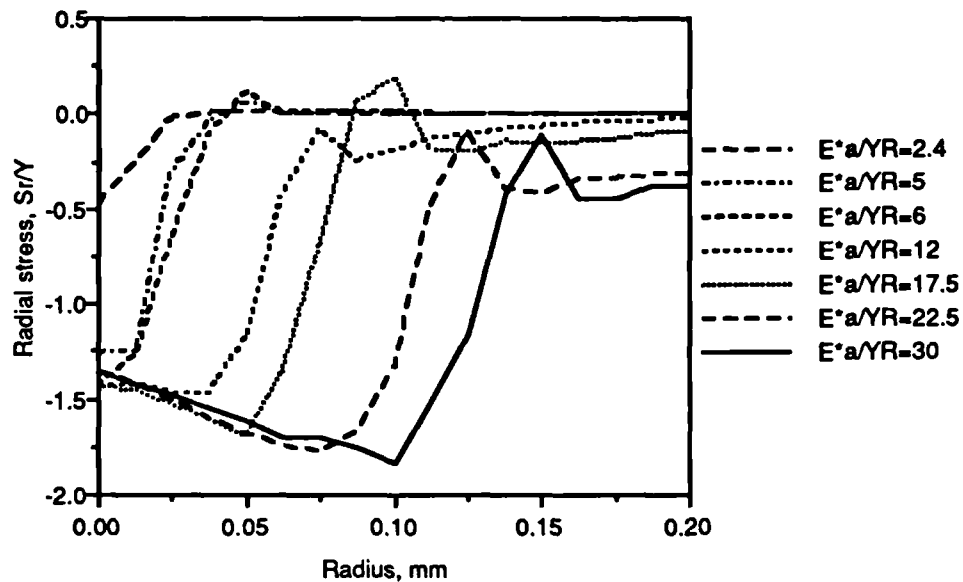


Figure 4.10. Radial stress profiles at the surface for seven steps during an elastic-plastic indentation by a rigid sphere.

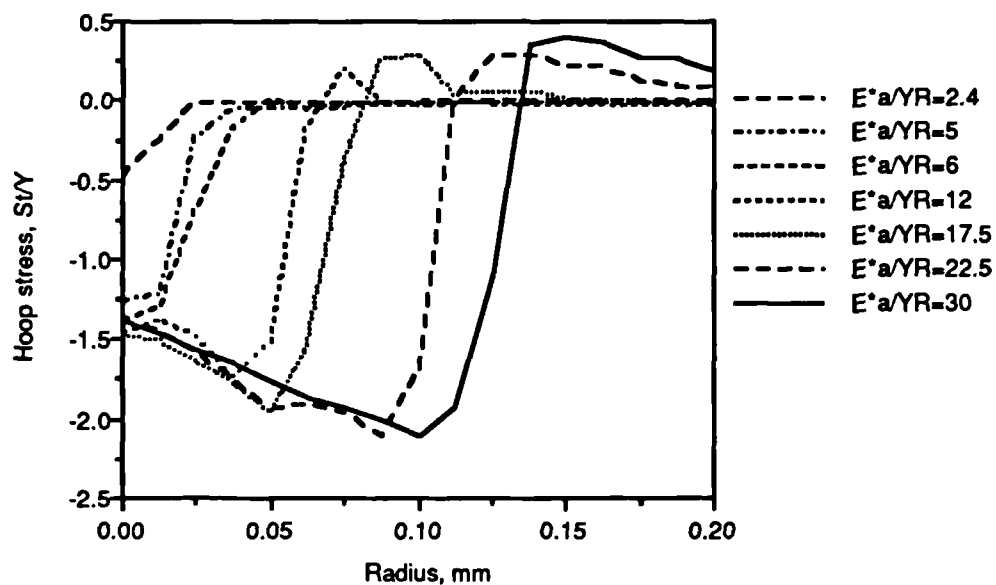


Figure 4.11. Hoop stress profiles at the surface for seven steps during an elastic-plastic indentation by a rigid sphere. Note, the tensile stresses developed at the edge of the contact.

Figures 4.12 and 4.13 show graphs of the development of the axial and hoop (which is equal to the radial) stresses down the axis of symmetry. In both cases the stresses are pushed further sub-surface as the indentation progresses. Tensile radial (and hoop) stresses are generated subsurface (at approximately $2a$ below the surface) under a region of high compressive stress. This region of subsurface tension can sometimes be the site for fatigue crack nucleation of shot peened components (which are in effect repeated spherical plastic indentations).

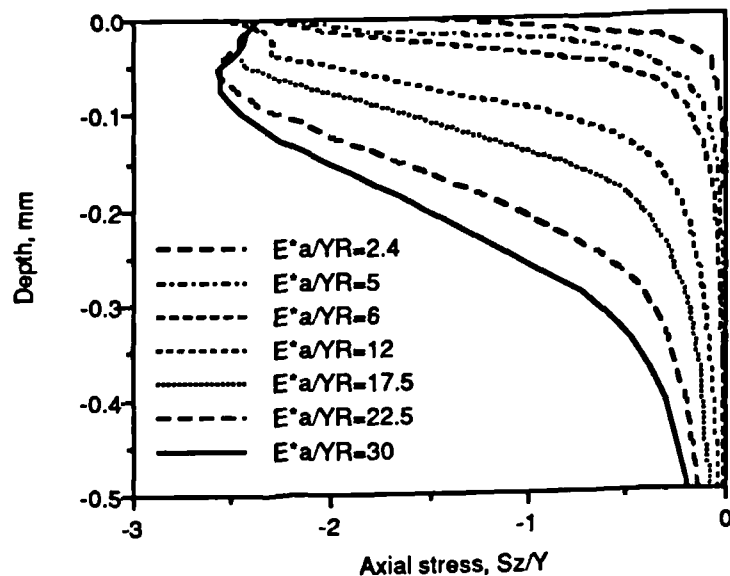


Figure 4.12. Axial stress profiles down the axis of symmetry for seven steps during an elastic-plastic indentation by a rigid sphere.

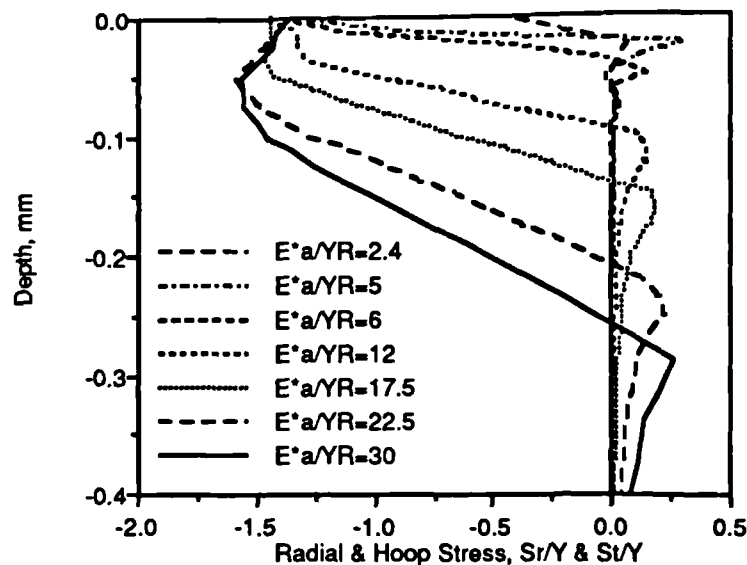


Figure 4.13. Radial (equals hoop) stress profiles down the axis of symmetry for seven steps during an elastic-plastic indentation by a rigid sphere.

Figure 4.14 shows the displaced shape of the surface during the indentation. The step change in the indentation slope is where the half-space and the rigid body come out of contact.

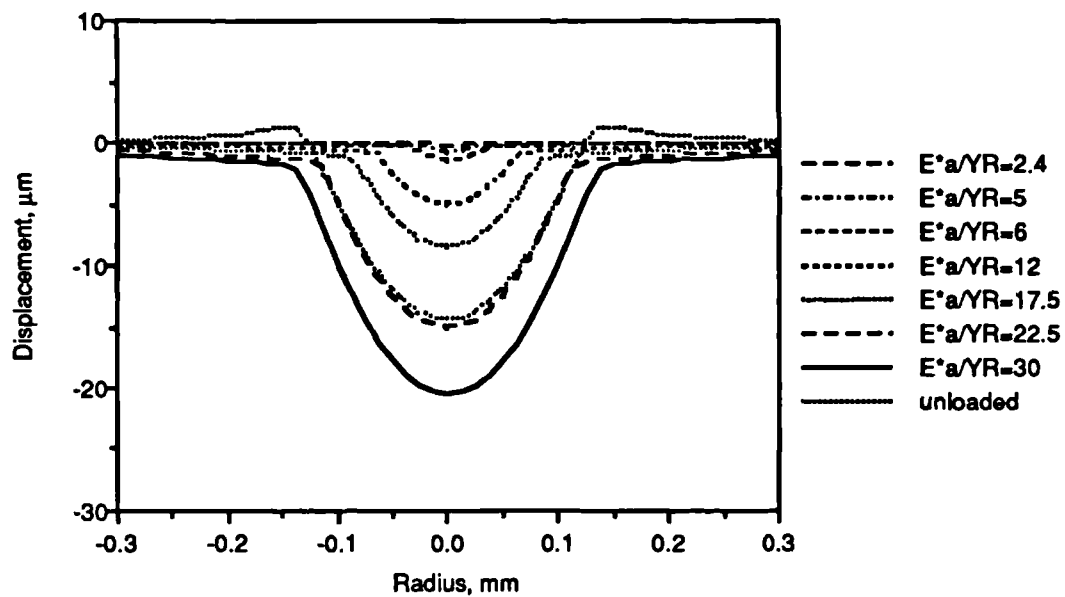


Figure 4.14. Displaced surface profile for seven steps during an elastic-plastic indentation by a rigid sphere.

The loaded indentation shapes are smooth; the surface remote from the contact region is also deflected. Unloading leads to elastic recovery; the depth of the dent is reduced by some 30%, the width remains fairly constant. The reasons for this and a discussion on the effects of work hardening are discussed in §4.4.3.

4.3.3. The Fully Plastic Regime

The last step in the loading cycle for the previous finite element mesh took the indentation into the fully plastic regime. The region of plasticity has reached the surface and the contact pressure profile is uniform; the non-dimensional strain E^*a/YR has a value of 30. At this point, it is instructive to examine the stress fields of this fully plastic indentation.

Figure 4.15 shows the distributions of radial, hoop, and axial stress along the surface of the half-space. The stresses are almost uniformly compressive in the contact area and fall rapidly at the edges. As discussed in the previous section, the hoop stress reaches a tensile maximum of about $0.5Y$ at the edges of the contact whilst the radial stress is compressive. Again, the small tensile axial stress visible at the edge of the contact is an erroneous feature of the finite element method caused by the interpolation routine.

Figure 4.16 shows the axial and radial stress distribution down the axis of symmetry. Along this axis the radial stress equals the hoop stress. The near surface stress state is compressive reducing at greater depth. The radial and hoop stresses become tensile, to a value of about $0.25Y$, at a position about $2a$ sub-surface.

The data compares well with the FE solutions of Sinclair and Follansbee (1985) and the Johnson cavity model (1970).

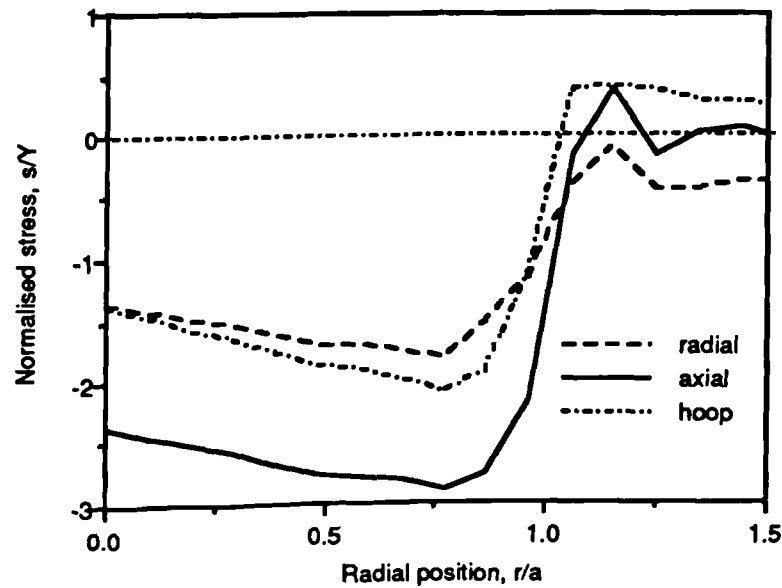


Figure 4.15 Radial, hoop, and axial stress distributions along the surface of a fully plastically indented half-space.

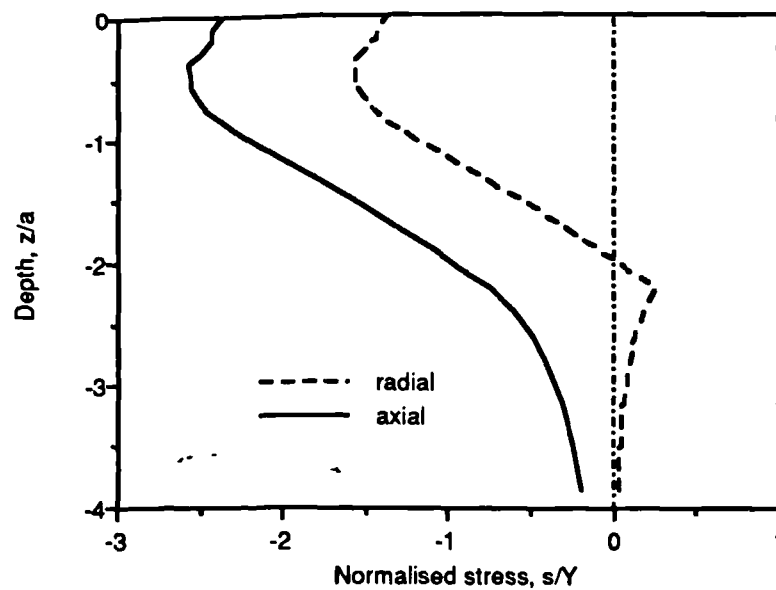


Figure 4.16. Radial (equals hoop) and axial stress distributions along the axis of symmetry of a fully plastically indented half-space.

The Figures 4.17, 4.18, 4.19, and 4.20 show contours of radial, shear, axial, and hoop stress within the elastic-plastic half-space (all stresses have been normalised by the yield stress). In each case only the data for the central region of the finite element mesh has been plotted. The plot represents only one side of the axis-symmetric half space. (i.e. the lower body as depicted in Figure 4.2)

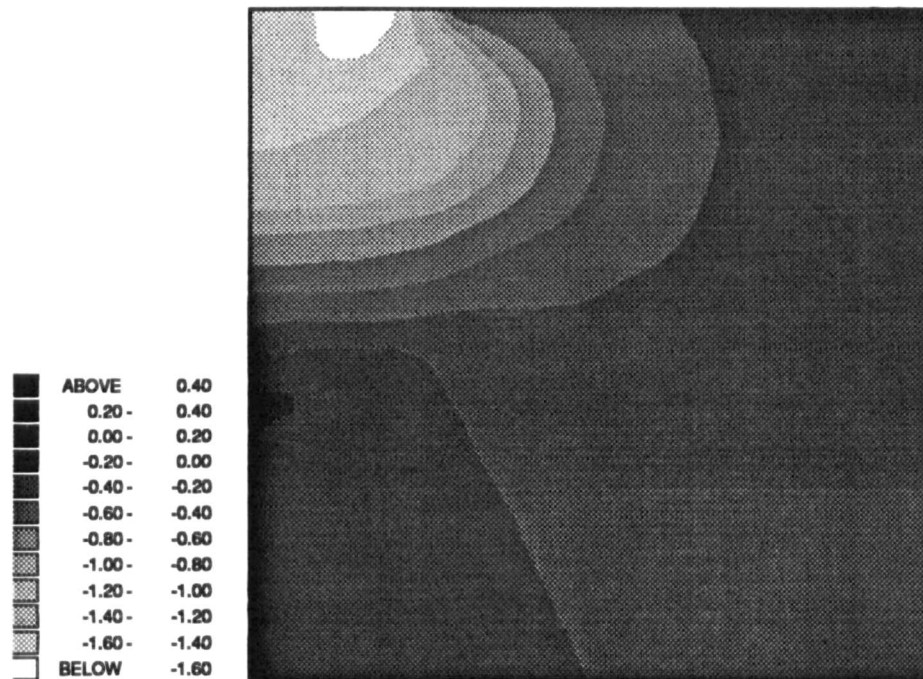


Figure 4.17. Contour plot of radial stress (normalised by the yield stress) stress below a fully plastically indented half-space.

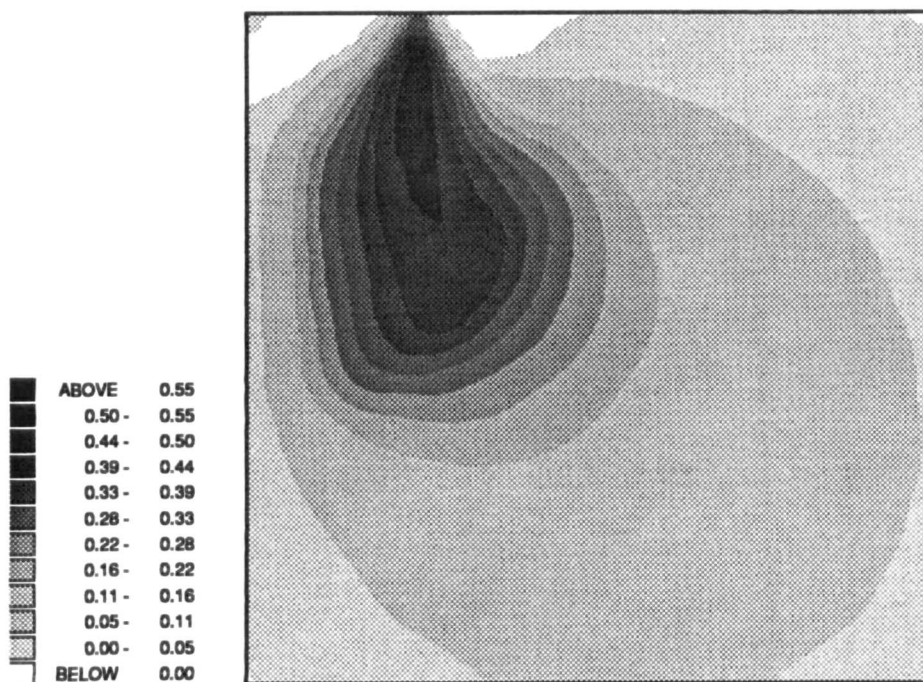


Figure 4.18. Contour plot of shear stress (normalised by the yield stress) below a fully plastically indented half-space.

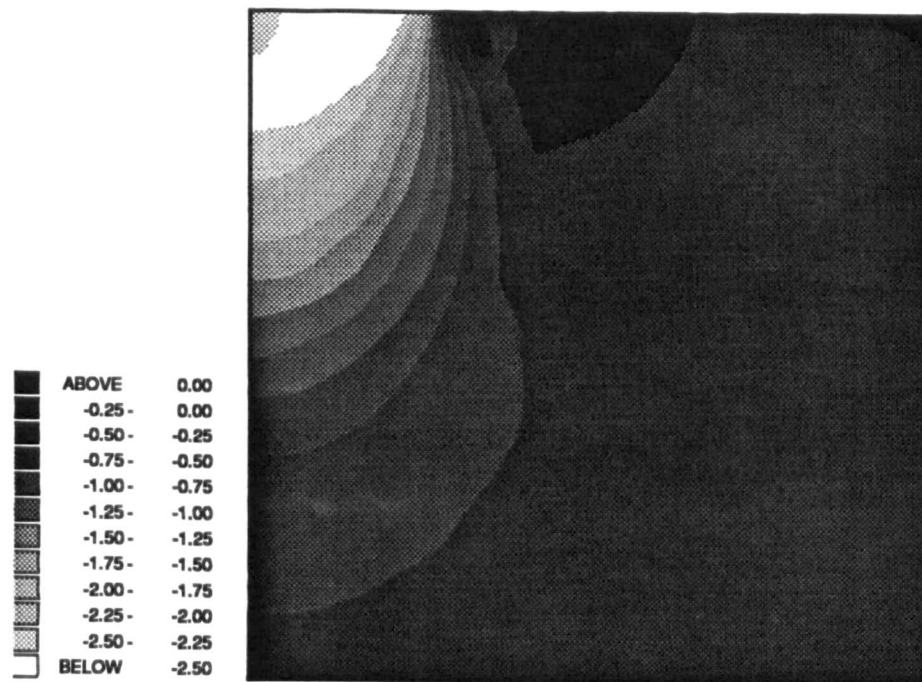


Figure 4.19. Contour plot of axial stress (normalised by the yield stress) below a fully plastically indented half-space.

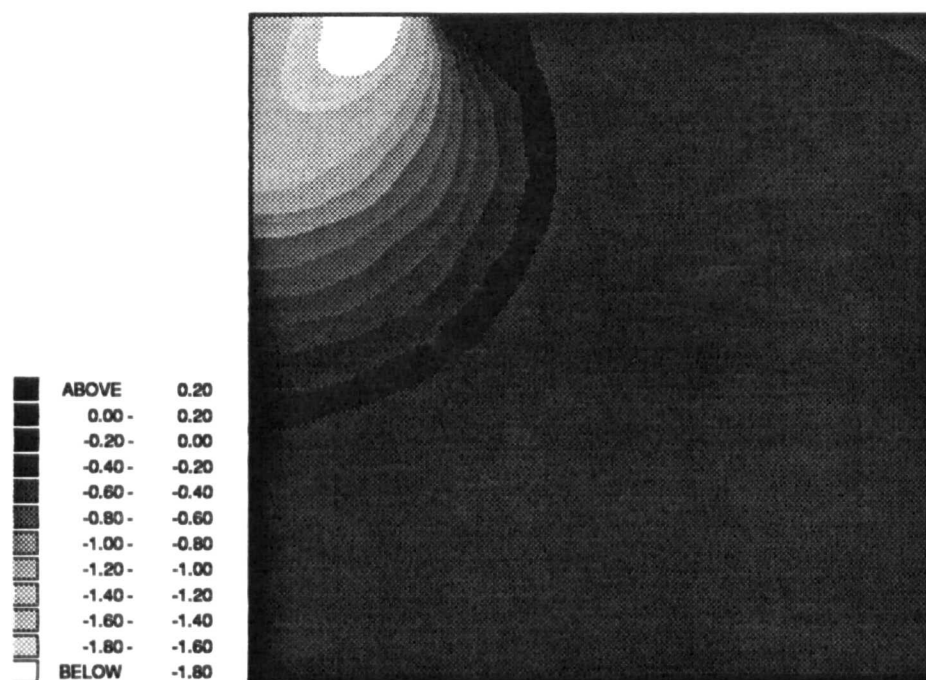


Figure 4.20. Contour plot of hoop stress (normalised by the yield stress) below a fully plastically indented half-space.

4.3.4. Residual Stresses

The final part of this finite element model was to include an unloading step. The applied load was removed and the body allowed to equilibrate; again the unloading process was subdivided into several increments to allow for path dependencies.

The pattern of residual stresses and strains were examined. Figure 4.21 and 4.22 show the distribution of residual radial and hoop stresses along the surface (the plotted radius is normalised using the contact half-width from the loaded step). Within the contact region the surface is in a state of radial and circumferential tension; with tensile stresses of approximately $0.3Y$. The hoop stress reaches a tensile peak of $0.8Y$ outside the contact region; whilst the radial stress becomes compressive.

The results of Johnson's (1968) experimental measurements of residual stress by scribing a fine mesh on the specimen surface and Olver's (1986) X-ray diffraction measurements are plotted on Figures 4.21 and 4.22 for comparison. The trends in both experimental analyses are similar to those from the finite element analysis.

Johnson (1985) describes a method for predicting the residual stresses in the contact region. The unloading process is equivalent to the subtraction of the elastic stresses. This assumes no further plastic flow occurs during unloading (i.e. the Baushinger effect is neglected, this is discussed further in §4.4.4). This analysis predicts radial and hoop tensile stresses of $0.4Y$ within the contact region.

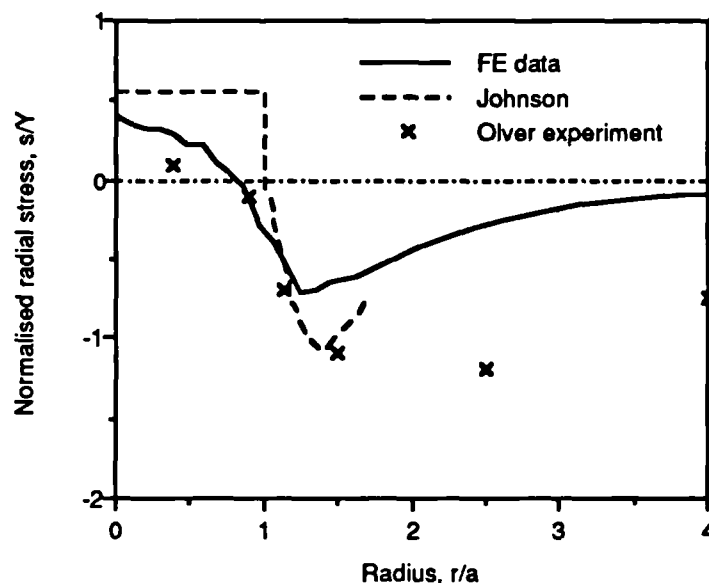


Figure 4.21 Residual radial stress distribution along the surface following a fully plastic spherical indentation. The results of the FE analysis are compared with measurements of Johnson (1968) and Olver (1986).

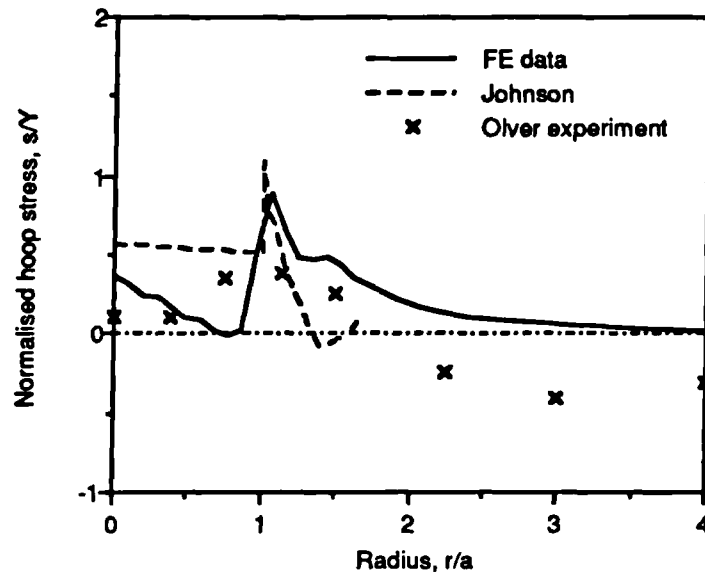


Figure 4.22. Residual hoop stress distribution along the surface following a fully plastic spherical indentation. The results of the FE analysis are compared with measurements of Johnson (1968) and Olver (1986).

This state of residual tension within the contact region (that was) occurs only in a thin layer near the surface. Figure 4.23 shows the distribution of the stresses down the axis. It can be seen that the stresses below the contact region become compressive at about $0.11a$.

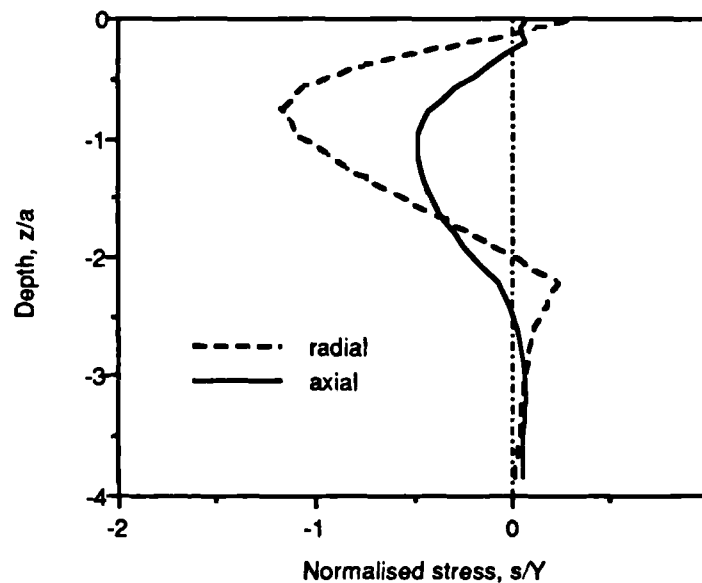


Figure 4.23 Residual radial (equals hoop stress) and axial stress distribution along the axis of symmetry.

Table 4.1 summarises the values of this radial surface tensile stress and depth with the work of other authors.

Author	Tensile residual stress (within contact)	Depth of tensile layer
This work	$0.3Y$	$0.11a$
Johnson (1968) experiment	$0.56Y$	$0.17a$
Johnson (1985) calculated	$0.4Y$	-
Olver (1986)	$0.1Y-0.3Y$	-
Sinclair & Follansbee (1985)	$0.35Y$	$0.10a$ to $0.13a$ ¹⁵

Table 4.1. Comparison of residual tensile stress magnitude and depth within the contact region from various sources.

The agreement is reasonably good considering the complex numerical processes used to achieve the final result. The finite element analysis at each load step introduces some numerical error; these errors tend to compound throughout the solution. At the unloading part of the analysis large elastic strains are subtracted from large total strains, which contain numerical compounded errors. This subtraction of these two large numbers to give the relatively small residual strains tends to leave the final result with a high percentage error. Indeed at higher levels of indentation (greater than about $E^*a/YR=50$) the finite element solution for the residual stresses was found to be unacceptably inaccurate.

The Figures 4.24, 4.25, 4.26, and 4.27 show contours of residual radial, shear, axial, and hoop stress within the elastic-plastic half-space (all stresses are normalised by the yield stress). Again, only one side of the axi-symmetric half space immediately below the indentation has been plotted.

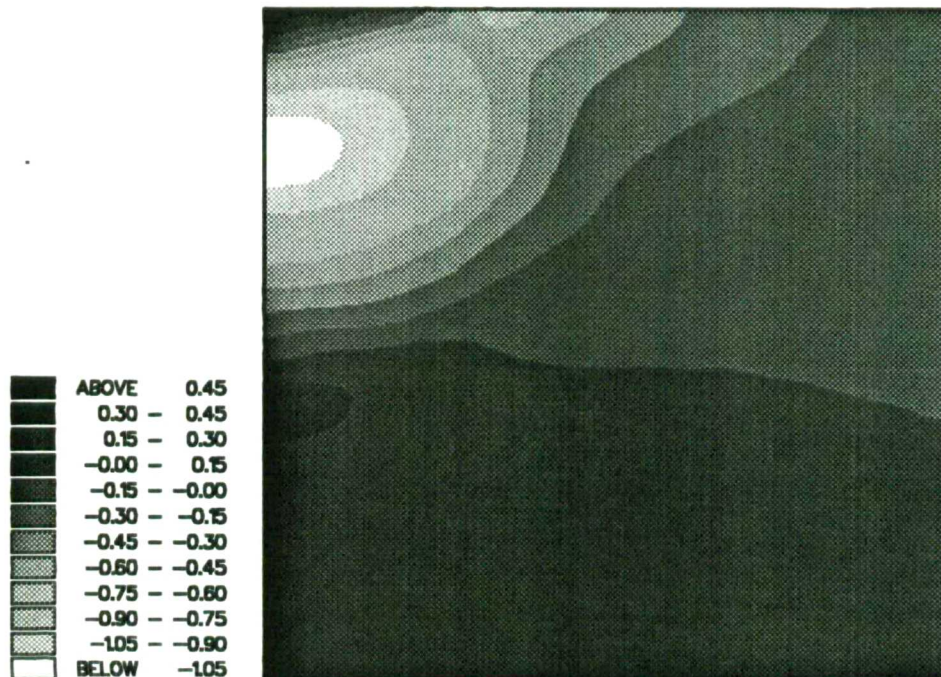


Figure 4.24. Contour plot of residual radial stress (normalised by the yield stress) below a fully plastically indented half-space.

¹⁵ It should be noted that the indentations modelled by Sinclair and Follansbee were much deeper ($E^*a/YR=170$); higher loading produced thinner tensile layers.

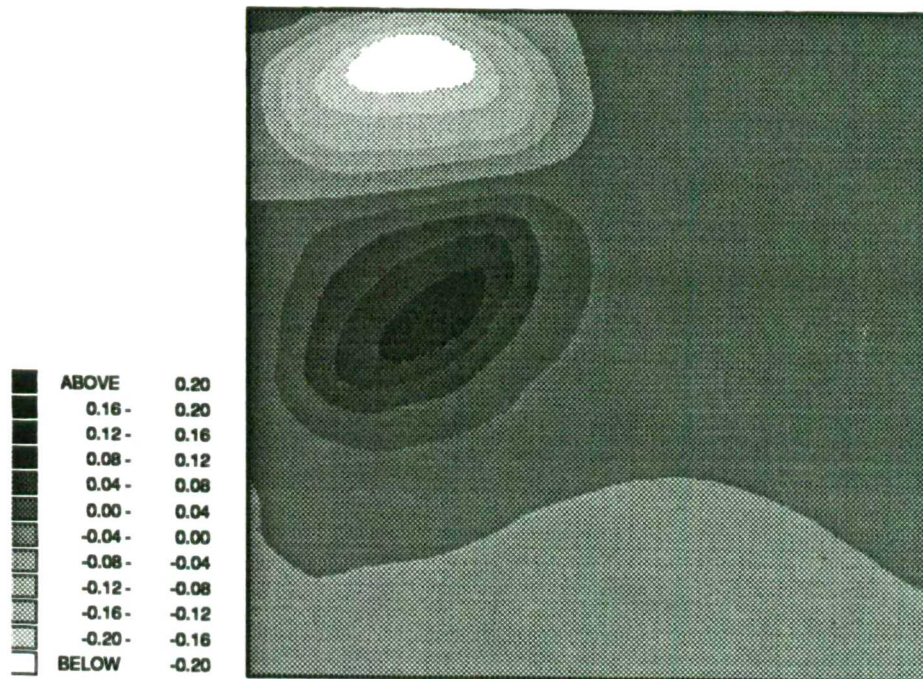


Figure 4.25. Contour plot of residual shear stress (normalised by the yield stress) below a fully plastically indented half-space.

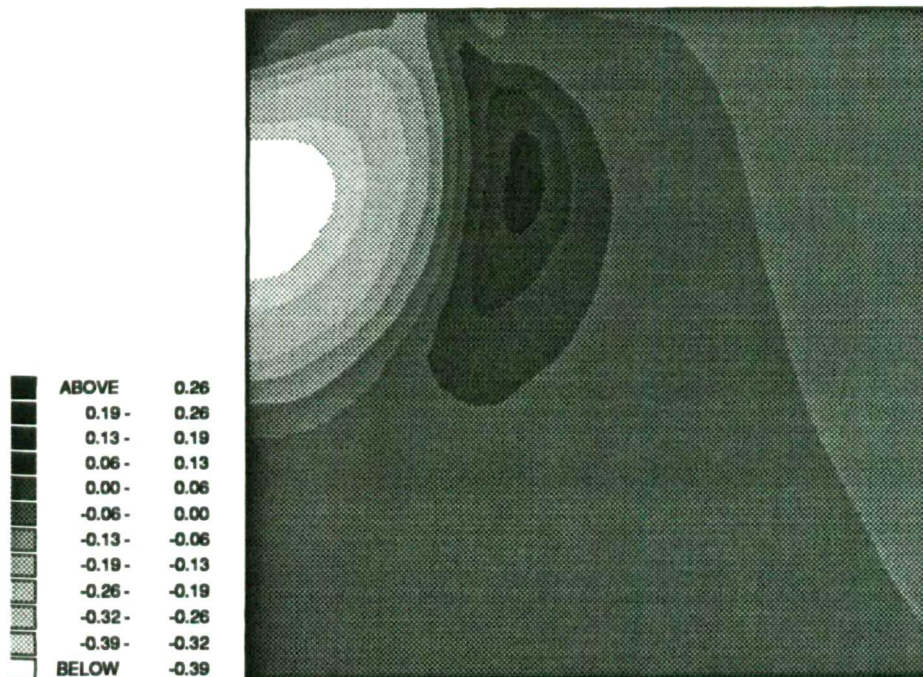


Figure 4.26. Contour plot of residual axial stress (normalised by the yield stress) below a fully plastically indented half-space.

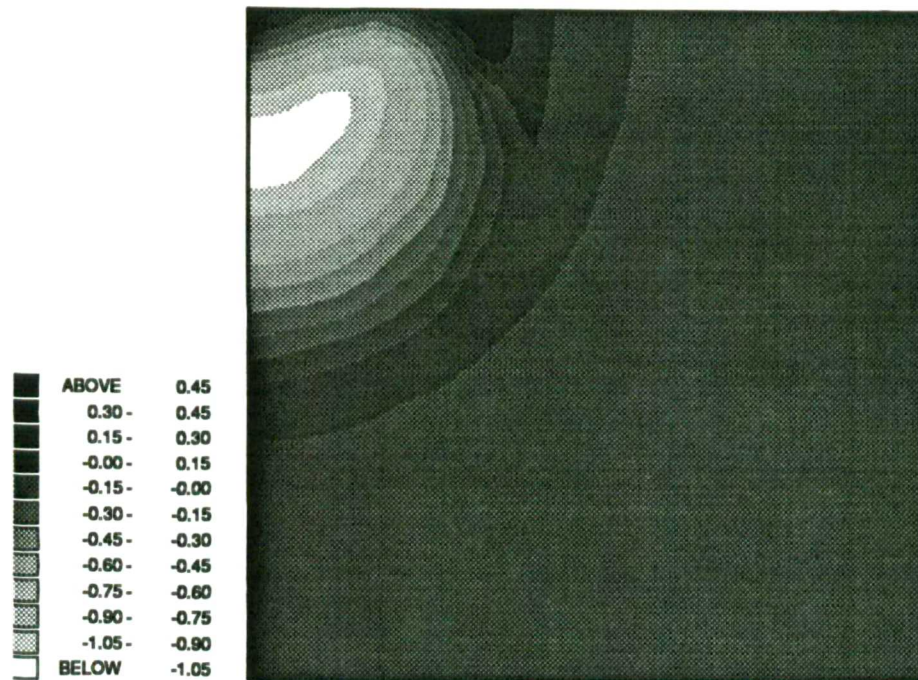


Figure 4.27. Contour plot of residual hoop stress (normalised by the yield stress) below a fully plastically indented half-space.

4.4. The Brinell Indentor

The work just described was for the indentation by a rigid body. The next part of the study was to model the indentations caused by pressing a tungsten carbide ball onto a bearing steel surface, and to compare the deformations with those from a Brinell hardness type test.

The analysis was also used to provide input for the fatigue life study. The results were then compared with the fatigue lives of bearings with inner raceways indented by tungsten carbide balls (described in the next chapter).

4.4.1. The Finite Element Tungsten Carbide Indentor

Figure 4.28 shows the mesh used. Both bodies were modelled using axi-symmetric eight noded quadrilateral reduced integration elements (CAX8R); the contact was defined by six noded axi-symmetric interface elements (INTER3A). The lower body was given the properties of a strain hardening bearing steel (Appendix 2 describes the determination of these properties using an indentation method developed by Tabor) whilst the upper spherical body was given those of tungsten carbide.

The base of the lower body was simply supported; as before, loads were applied to the corner node of the upper body, the top face being constrained to move in a horizontal line with this corner node. The load was applied over several steps; residual dent shapes were obtained by restarting the analysis at one of the previous loading steps and following with an unloading step.

Again elastic stresses, elastic-plastic deformation and full plasticity gave good results; in general agreement with the previous model and the analysis of other authors. In this section it is the size

of the indent which is of interest; the stress fields during the indentation are similar to those described in §4.3.

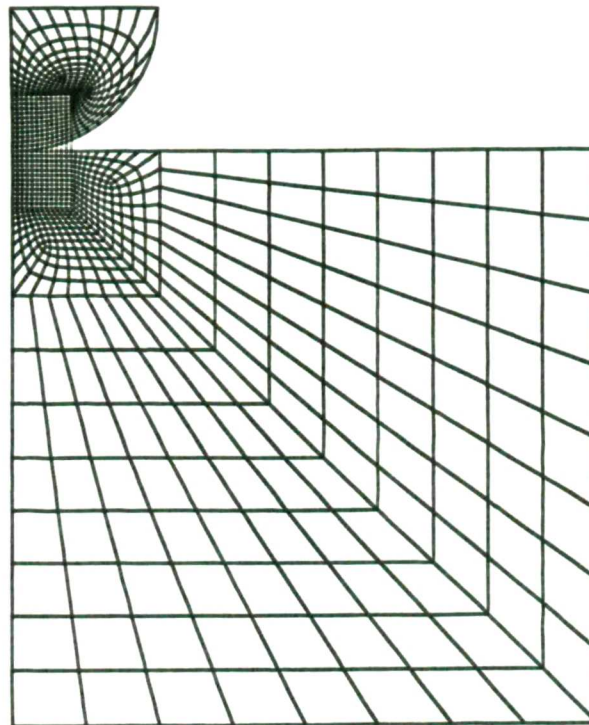


Figure 4.28. Finite element mesh used in the modelling of tungsten carbide spherical indentation.

Figure 4.29 shows the residual dent shapes generated by this analysis by the application of five different loads. The dent shapes are geometrically similar; little material is built up at the shoulders of the dent.

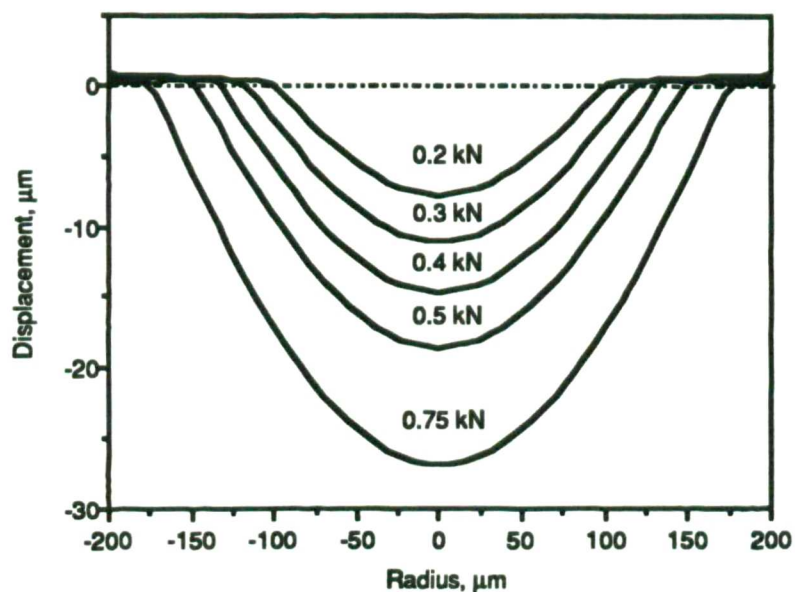


Figure 4.29. Residual dent shapes generated by five different loads in applied to the finite element mesh.

4.4.2. The Indentation Experiments

For comparison some simple indentation experiments were performed. A 10mm diameter tungsten carbide ball was mounted in a press between two well oiled bearing steel platens (the elements from a spherical roller bearing were convenient specimens). The load was applied slowly and then released. Dent depths and widths were measured using a Form Talysurf.

Table 4.2 is a shows the residual dent data from the finite element study compared alongside the experimental data (note the ball indents have been scaled down by a factor of 10 since FE studies were performed using an indenter of diameter 1mm). The extent of the indent is measured by the parameter a/R where a is the dent semi-width and R is the radius of the indenter.

Applied Load, kN	a/R (FE data)	FE Depth, μm	Experiment Depth, μm	FE Width, μm	Experiment Width, μm
0.2	0.20	7.8	5.1	186	198
0.3	0.24	11.1	8.2	218	222
0.4	0.26	14.7	11.6	250	263
0.5	0.30	18.6	14.2	282	288
0.75	0.35	26.8	20.7	344	340

Table 4.2. Comparison of indentation sizes for FE model and experimental data (experimental indents scaled down by a factor of 10)

The form of the dents was similar to those generated by the finite element study; again with little evidence of built up shoulders. The finite element model generally under predicts dent depths by about 25%; the dent widths correspond closely to experimental data. This error may, in part, be a result of the uncertainty of strain hardening properties (the following section develops this idea and generates some comparative data).

Little elastic recovery occurs from the sides of the dent (in this region stresses are tensile and little recovery would be expected). The finite element analysis shows 30-40% elastic recovery of the dent on unloading. The stresses below the contact are high compressive and elastic unloading will therefore lead to significant decrease in strain, and the dent shallows.

4.4.3. Effects of Strain Hardening

ABAQUS has the facility for modelling work hardening material. As input, the program merely requires a linear approximation to the post yield stress-strain relationship.

Data was not available for the strain hardening of typical bearing steels. A materials test would require the machining and heat treating of a torsion test specimen. This time consuming process was avoided by obtaining the required data using a Brinell indenter in a method described by Tabor (1951). Appendix 2 contains a description of the experiments.

The finite element indenter analysis was re-run using isotropic and kinematic work hardening properties (see §4.4.3 for definitions). Figure 4.30 compares the dent shapes from the purely plastic, isotropic, and kinematic work hardening assumptions with a Talysurf trace from an experimental indentation.

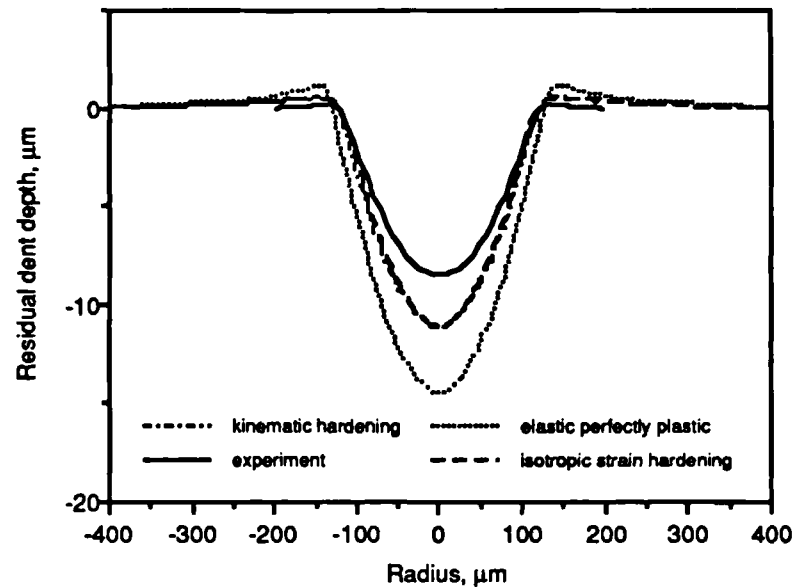


Figure 4.30. Dent shapes from finite element analysis of indentation of purely plastic, isotropic, and kinematic work hardening materials; compared with a Talysurf trace from an experimental indentation.

The use of a strain hardening model reduces displacements by about 15% relative to an elastic perfectly plastic analysis. However, the predicted displacements still exceed experimental measurements.

The elastic perfectly plastic analysis has developed higher dent shoulders than the work hardening analyses. This 'piling-up' was also noted by Tabor (1951) from his experimental observations. Elastic perfectly plastic materials tend to produce shoulders while work hardening materials tend to cause a 'sinking-in'.

Figure 4.31 shows the reason for this effect. A material which has been heavily work hardened (like bearing steel) generally shows little further strain hardening and is close to an elastic-perfectly plastic material. In the indentation process first yield occurs in a region sub-surface below the indenter. Little strain hardening occurs and the yield process carries on in this region. The material displaced then moves to the free edge close to the contact; and shoulders are formed (as in Figure 4.31a).

However, for a material in its annealed state strain hardening can occur. So once yield has occurred during an indentation, the material in the yielded region strain hardens and it becomes easier to displace material deeper below. This displaced material will then flow to a region further from the contact; the result is 'sinking-in' (as in Figure 4.31b).

The modelling of work hardening reduces the depth indentation considerably. The use of a, generally more realistic, kinematic strain hardening law makes little difference to the use of a simpler isotropic law. This suggests that the Baushinger effect seems to make little difference (this is examined further in the next section). Both models predict deeper dents than those found in the experiment; this may be a result of the uncertainty of the work hardening properties.

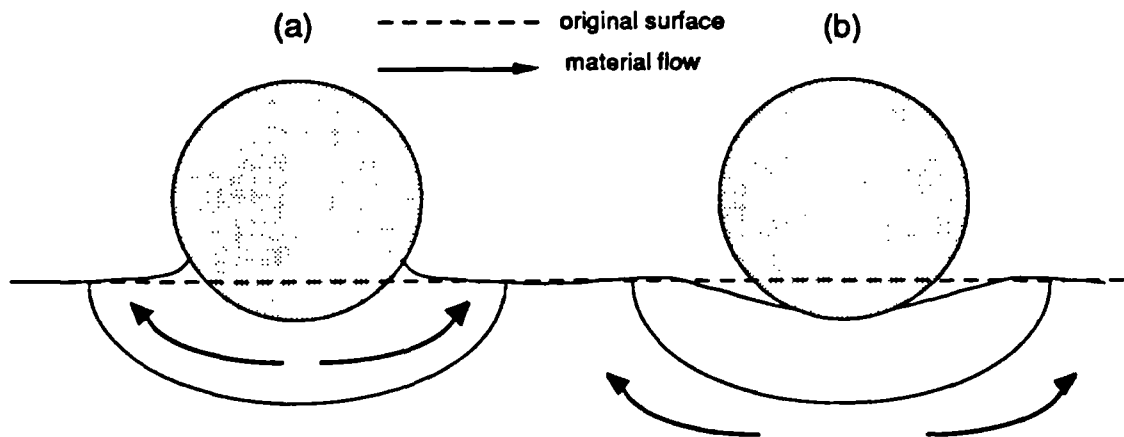


Figure 4.31. Diagram of the indentation of (a) elastic perfectly plastic (highly worked and therefore unlikely to strain harden) material and (b) annealed material (with the ability to strain harden). Showing flow of displaced material and the resulting piling-up or sinking-in.

4.4.4. The Baushinger Effect and Yield During Unloading

A material which strain hardens isotropically is governed by a yield locus which expands in all directions as a result of plastic deformation. An alternative and more realistic model of strain hardening is the linear kinematic model; which states that the yield locus is of constant size but free to translate in space. When compressive plastic strains build up the tensile yield stress is reduced, and vice versa. This process is called the Baushinger effect, and implies that yield can occur during an unloading process. Figure 4.32 diagrammatically explains these two work hardening models.

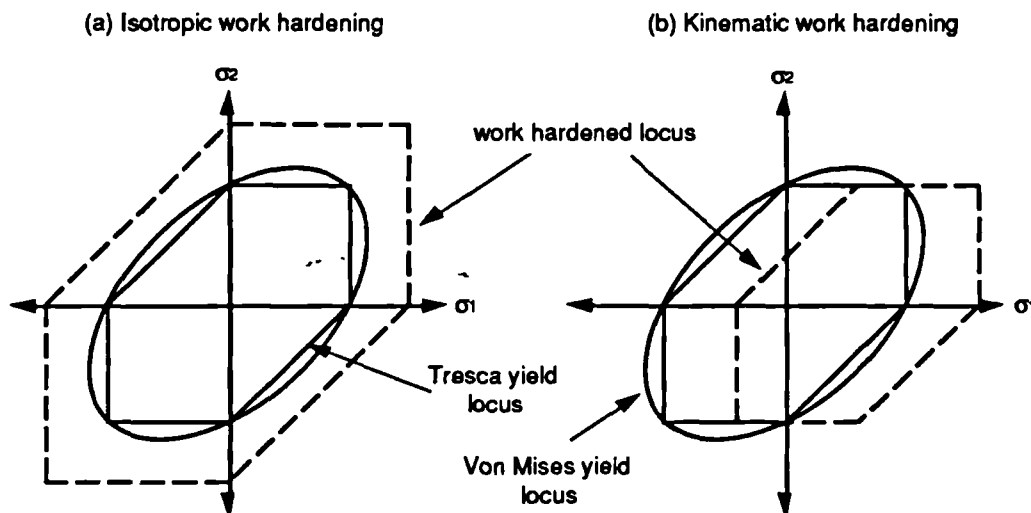


Figure 4.32. The yield locus of materials which harden (a) isotropically and (b) kinematically. In the former the locus expands in all directions; so yield in compression increases the yield stress in tension. In the latter the locus is of fixed size and shifts with loading; so yield in tension reduces the yield stress in compression.

The finite element model described above was used to investigate whether this yield during unloading occurs. The plastic strains were recorded before and after the unloading step of the analysis. Figure 4.33 shows contours of the increase in equivalent plastic strain (which indicates whether yield has occurred) during the unloading step.

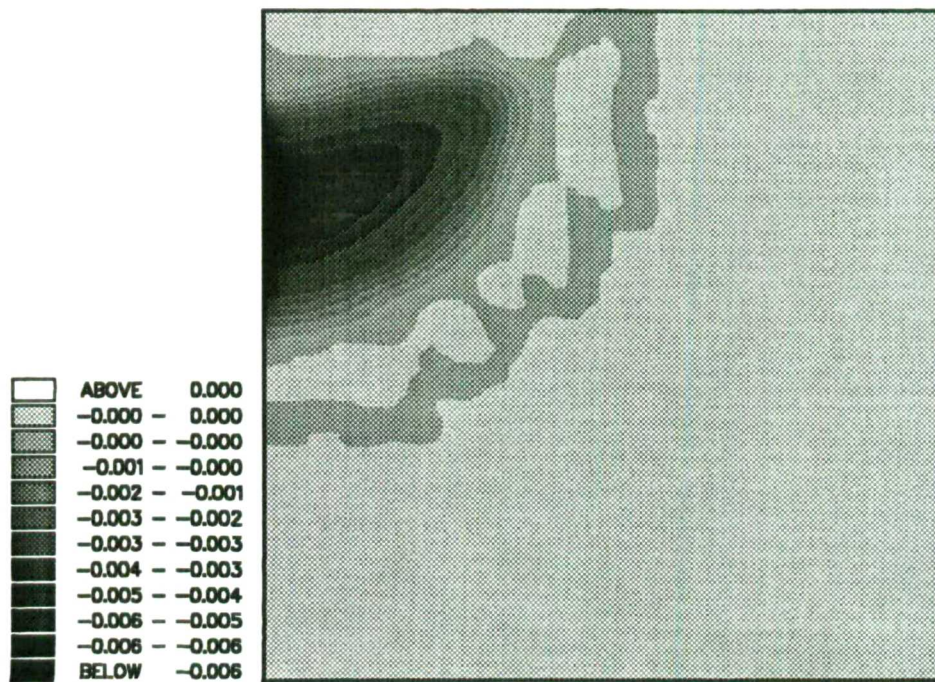


Figure 4.33. Change in equivalent plastic strain during the unloading of a spherical indentation. This demonstrates that further plastic flow occurs during unloading. (Again, only one side of the axi-symmetric half space is shown as in Figure 4.2).

Clearly, further plastic flow occurs during the unloading step. This is confined to the subsurface region; because this was where the majority of the plastic flow occurred in the loading steps (i.e. the yield locus was shifted further in this region, therefore the stresses in the opposite sense applied during unloading exceeded the now reduced yield point). However, the previous section demonstrated that this confined plastic flow had little effect on the size of the resulting dent.

4.5. Conclusions

Finite element analysis provides a flexible method for the modelling of surface indentation. The general purpose code ABAQUS has been used to model indentation by rigid or hard spherical bodies. The modelling of ductile particles being overrolled in a contact is not straightforward; gross plastic flow makes the finite element analysis lengthy. The sort of analysis performed here is more like the overrolling of brittle materials once they have been fractured to a ultimate size; they then behave as very hard indentors.

Several finite element meshes were developed, the results from elastic analyses were found to be very close to the Hertzian solutions. Elastic-plastic transition and fully plastic stresses were also studied and found to be comparable with those from experimental and theoretical studies of other workers.

Unloading of the indentors gave residual stress patterns and resulting indentation shapes; these were also found to compare well with other studies. The modelling of a strain hardening material was found to reduce the size of the resulting indents, but not to the extent necessary to agree with experimental data.

The residual stress fields and deformed shapes predicted in this analysis were used in conjunction with a fatigue life model; this work is described in the next chapter.

CHAPTER 5

FATIGUE LIFE MODELLING

5.1. Introduction

The effect of an indentation in a rolling contact is to act as a stress raiser from which fatigue cracks may initiate. The need for this project developed from the problems experienced by gear box manufacturers whose rolling bearings were suffering from premature failure. Failures were caused by surface fatigue initiated at debris damage sites. Figure 5.1 shows a fatigue spall initiated from such a debris dent. The dent is located at the trailing edge of the fatigue spall.

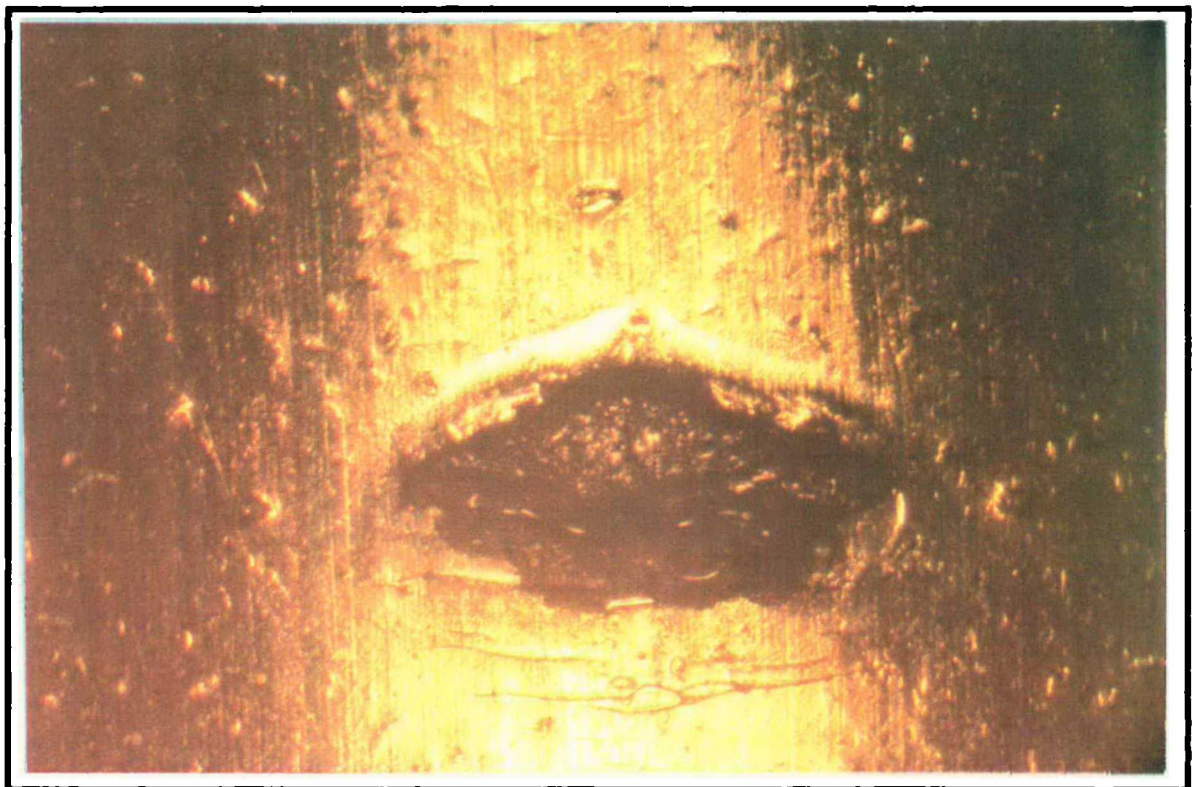


Figure 5.1. Photograph of rolling bearing fatigue spall initiated at a surface debris dent (overrolling direction top to bottom).

The next sections describe how the finite element modelling of indentation can be used in conjunction with a rolling contact fatigue model to predict reduction in the contact life of a damaged surface. The work was done in conjunction with co-workers at SKF ERC and has been published in three papers (Dwyer-Joyce et al 1991, Beghini et al 1992, and Lubrecht et al 1991).

Three separate studies are described, which model the effects on fatigue life of;

- (i) Axi-symmetric indentations by a tungsten carbide ball.
- (ii) Plane strain indentations by a ridge.
- (iii) Plane strain asperity flattening

The work presented in this thesis is not intended as a rigorous study; but rather as an introduction into how surface dents reduce fatigue life and a discussion of the parameters which are significant.

5.1.1. Rolling Contact Fatigue

The process of rolling contact fatigue is characterised by the initiation of surface or sub-surface micro-cracks at localised stress concentrations by the action of cyclical stressing. These micro-cracks then rapidly propagate to form spalls or pits. The presence of these pits, on the contacting elements, leads to massively increased vibration.

Way (1935) performed some of the first experiments into rolling contact fatigue, using rollers lubricated with various oils. He noted the formation of fatigue pits, and showed that pitting could be reduced by using high viscosity oils and rollers with hard polished surfaces.

Many attempts have been made to isolate the stress components responsible for the initiation and propagation of fatigue cracks, criteria based on equivalent stress, maximum shear stress, and maximum orthogonal shear stress have been suggested. Popinceau (1981) provides a good review.

Lundberg and Palmgren (1947) postulated that the fatigue life of a rolling element bearing was related to the applied load and its load carrying capacity. They suggested a probabilistic relationship for the prediction of fatigue life;

$$\text{Log} \frac{1}{S} = K N^e V \frac{\tau_0^c}{z_0^h} \quad (5.1)$$

where S is the cumulative probability of survival through N cycles, τ_0 is the maximum orthogonal shear which occurs at a distance z_0 below the surface, and V is the stressed volume. The exponents c , h , and e are constants greater than zero. This relationship has been adopted as a standard for the calculation of bearing lives.

An adaptation of this analysis was made by Ioannides and Harris (1985). Their model used an elemental discretisation of the stressed volume; such that individual elements may contain defects leading to a higher probability of failure. The fatigue life is then a summation of all the elemental lives. The fatigue stress formulation included a fatigue life threshold which varied with shear stress amplitude. A hydrostatic pressure component was added by Ioannides et al

(1988). A tensile hydrostatic pressure opens fatigue cracks and thus reduces the life integral whilst compressive stresses close cracks and increase the value.

This discretisation of the stressed volume, in this manner, allowed for local stress raising effects to be considered. So if the stresses and displacements generated from a debris indentation can be modelled they can then be superimposed on the normal overrolling stresses and used in the fatigue life calculation.

5.1.2. Debris Fatigue In Rolling Bearings - Experimental Data Review

Yardley & Crump (1969), in a paper on observations of bearing failures, discussed the effects of lubricating grease contamination. Bearings were observed with pitting marks and possible resulting fatigue spalls, and wear rims on roller ends; these effects were suggested to be caused by dirt introduction during grease re-packing.

Dalal & Senholzi (1976) performed some novel experiments where Vickers hardness indentations were made on rolling bearing inner raceways. The bearings were then run to failure, with resulting life reductions of about seven times; although some bearings failed at locations other than the indentation marks. The authors discuss the fatigue process in relation to the direction of sliding in the rolling elements; with the dent deforming in the direction of sliding and inducing a spall. The resulting wear debris and surface damage were also studied; the process of debris overrolling and flattening, to a 'safe' shape, were highlighted.

Sugiura et al (1981) suggested a slightly different approach; instead of filtration and control of particle size, they heat treated the bearing materials to reduce 'crack sensitivity'. The bearings with optimum heat treatment had two to three times the fatigue life under clean lubrication and five to six times under contaminated conditions. Various particulate contaminants were tested for their ability to initiate cracks; it was found that large hard particles were the most detrimental

Loewenthal, Moyer, & Needelman (1982) carried out further test to evaluate the effect of filtration on bearing life. An 'ultra-clean' lubrication environment, with 3 μ m barrier filtration, was used as a base line. At coarser filtration levels (49 μ m) lives were reduced by a factor of three, whilst Weibull slopes increased.

Sayles & Macpherson (1981) presented a study on the effects of real gearbox debris in bearing life. The contaminated oil was pumped through various rating cartridge filters and used to lubricate a pair of cylindrical roller bearings. They found that lubricant film thickness had little effect on bearing life, but the filter rating had a much larger influence (seven fold increases in life were noted with fine filtration). A most interesting finding was that damaged surfaces did not recover when run with cleaned lubricant; it was suggested that the damage caused during the first cycles was responsible for early failures. In keeping with previous work, it was found that Weibull slopes increased as filtration coarsened; the authors suggested that this was the result of the convergence seen at high stress levels on stress/cycles curves. Metallurgical investigations suggested much of the early failure was due to surface-initiated damage. SEM investigation of the damage raceways (by Bachu 1980) showed greater damage with the coarser filtration tests. Webster (1986) later 3D mapped some of these dents and found shallow rounded dents up to 5 μ m deep and 300 μ m in diameter.

Lorosch (1983) described experiments where artificial dents were introduced onto the inner raceway of test ball bearings. The author noted that indentation slope had a strong influence on fatigue lives (life reduction factors of 400 were recorded for some deep, steep slope indentations. Investigation of the fatigue spalls showed crack nucleation occurred at the trailing

edge of the dent near the built-up indentation shoulder. In addition to these tests Lorosch pre-damaged bearings with debris particles (file chips, sand, corundum, and grinding chips) suspended in a grease. The inner raceways were then removed and completed with fresh outer raceways and balls and endurance tested under the clean conditions. Again failure occurred at the pre-damaged locations with life reductions of about 140 times (for corundum debris). In further experiments various size bearings were artificially indented and endurance tested; bearings of small contact ellipse were found to be particularly sensitive to contaminants. The plotted graphs of relative life against indentation size are of particular note; the results indicated that shallow low slope dents have the least effect on fatigue failure and there appeared to be a size threshold where the indentation has no effect on fatigue failure.

5.1.3. Debris Fatigue In Rolling Bearings - Theoretical Model Review

Webster et al (1985) used the Ioannides and Harris (1985) life model to study the effects of debris denting. Real debris dents were mapped and the topography input into a surface contact model to represent indent overrolling; the resulting contact pressures were then applied to a finite element mesh and sub-surface stresses found. These stresses were then used in the fatigue life formulation to obtain risk maps and life reduction factors. Areas of high risk were found around the dent shoulders (caused by the steep slopes) and in the sub-surface areas of maximum shear stress (caused by the Hertzian overrolling).

Lubrecht et al (1990) extended the analysis further by including calculations based on a lubricated contact. The oil film generally damps pressures and gradients are reduced. The authors found higher pressures at the dent trailing edge in transient calculations; these would induce higher stresses and so provide a rationale for this being the spall initiation site, as recorded by previous experimenters. Life calculations were compared with the dry contact assumptions and found to be only 6% different.

One of the limitations of the work of Webster et al (1985) was that the residual stresses from debris indentation were neglected. Hamer et al (1988) used the slip line field method to remedy this. A rigid particle was modelled indenting a plastic half-space; the residual stresses were then superposed on the overrolling stresses in the fatigue model for life predictions. It was noted that expected fatigue lives may not increase as rapidly as would be predicted with decreasing load; and that the lives were very sensitive to dent size in relation to roller radius.

Ko and Ioannides (1988) used pressure profiles from Hamer et al (1987) and applied these to a finite element mesh; unloading giving the residual stresses. Again the fatigue life model was employed to determine contact life.

The approaches of these last two papers highlight some limitations in the modelling of indentation fatigue relationships. The slip line field analysis deals solely with plastic indentation. The elastic stresses during indentation cannot, particularly with shallow small size dents, generally be neglected. The finite element approach using a contact pressure to model elastic-plastic indentation is a very broad assumption; the flat pressure profiles generated during a fully plastic indentation can not be applied to produce an indentation (since plastic flow is path dependent). By using an elastic-plastic finite element model of two contacting bodies these inaccuracies have been overcome. The approach is described in this chapter.

5.2. Finite Element Indentation and Fatigue Life Calculation

The finite element method can provide useful information in the study of damage initiated rolling contact fatigue. Deformed shapes following indentations (similar to those described in §4.4) and residual stresses (as described in §4.3.4) can be used in conjunction with a rolling contact fatigue model to predict contact life.

5.2.1. The Overrolling of an Indent

When a surface dent is present on a contacting component (a rolling element bearing or a gear tooth) the stress field during the overrolling process is altered. The contact pressure rises over the shoulders of a dent and subsequently falls (to zero if the dent is deep and the bodies come out of contact). These steep pressure gradients induce high subsurface shear stresses beneath the dent shoulders.

Webster (1986a) developed a numerical dry contact model to predict contact pressures when a smooth cylinder is loaded onto a real rough surface. The model uses digitised surface measurements from a profilometer. The analysis divides the rough surface into elements (typical of $2\mu\text{m}$ width) over which the pressure is constant. Using elasticity equations the displacements due to each of these pressures can be calculated and superposed. By simultaneous solution of the contact pressures and displacements, the approach of the two surfaces can be adjusted until the integrated contact pressure equals the applied load. The final contact pressure distribution for a given load and the deformed shape of the profile are given as output. Sliding friction and the transfer of shear stresses can be included but other effects of lubrication are not considered.

Webster used this contact model to predict elastic contact pressures produced by the overrolling of a real dent. Figure 5.2 shows such a sequence of pressure profiles.

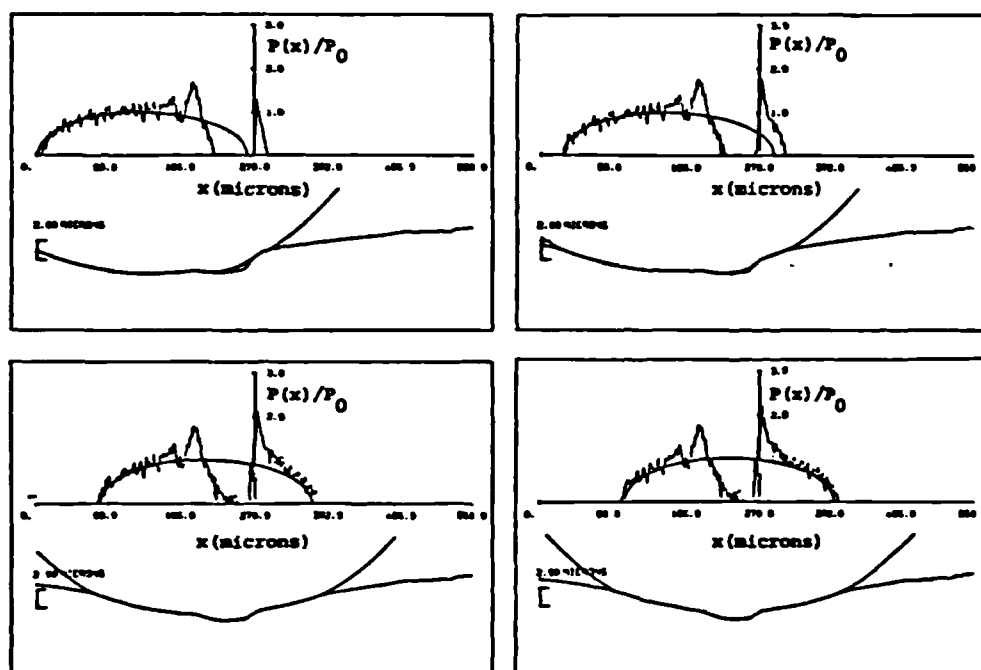


Figure 5.2. Contact pressures generated by the overrolling of a real debris dent; obtained from the Webster (1986a) contact model.

These contact pressures can then be used to calculate subsurface stresses. Webster used a finite element method; a more satisfactory solution is to sum the subsurface stresses resulting from each individual uniform pressure element.

A pattern of subsurface stresses is generated for each position of the contacting body. A single element in the mesh undergoes shear stress in one direction, which falls to zero as the body is directly above, and then rises to a peak in the other direction. This alternating pattern of stresses is superimposed on the stresses caused by the overrolling of the indent. Figure 5.3 shows a schematic diagram of the overrolling process and the regions of high shear stress caused by the cylinder at its central position.

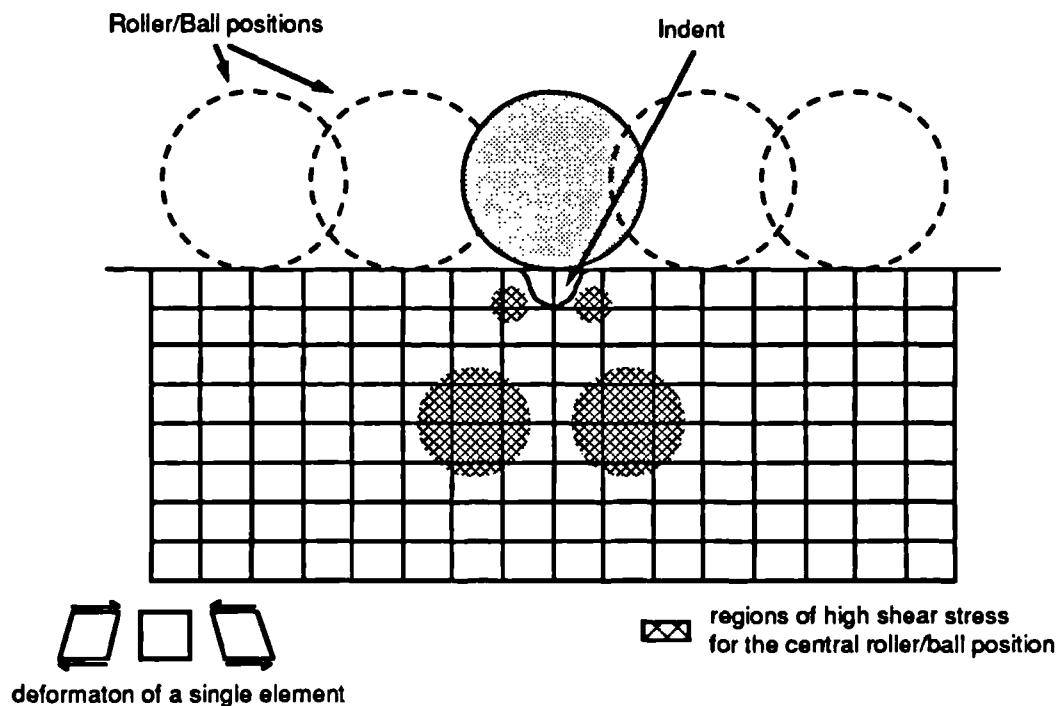


Figure 5.3. Schematic diagram of the overrolling process as modelled by the elastic contact model (inset shows the deformation of a single element and how the shear stress alternates during overrolling).

The method may then be used to predict subsurface elastic stresses caused by the contact of a smooth curved body and an indented surface. The static analysis may then be repeated for a range of cylinder locations to give the stress variation during overrolling. It is this data which is used in the calculation of rolling contact fatigue life.

5.2.2. The Fatigue Life Formulation

The displaced surface profile of a dent created using the finite element indentation is retained and used with the above contact model to give subsurface stresses for each roller location. The residual stresses are extracted from the finite element analysis. The FE meshes are not on a regular grid; the mesh is finer in regions of high stress gradient. The stresses at the nodal points were interpolated onto the regular contact model xy grid (the refinement of this grid was equal to or less than the smallest element of the FE mesh; in this way no stress data was lost). The

residual stresses (now interpolated onto a regular grid of the same size as the contact model grid) are superimposed on the subsurface overrolling stresses at each of the contact positions.

The fatigue life formulation used is a development of the Ioannides and Harris (1985) model where the probability of survival, S is given by;

$$\ln\left(\frac{1}{S}\right) = N^e \int_V \frac{(\tau_a - \tau_u)^c}{z^h} dV \quad (5.2)$$

Simplifications of the model (see Lubrecht et al (1990) for a full explanation) have changed the coefficients c , e , and h and removed the depth weighting z , giving a life L ;

$$L^{-\frac{10}{9}} \propto \sum (\tau_a - \tau_u)^{\frac{31}{3}} \delta V \quad (5.3)$$

where τ_a is the maximum value of the shear stress amplitude corrected by the hydrostatic pressure at any angle direction θ and as given by;

$$\tau_a = \max(\tau_\theta - 0.3 p_H)_a \quad (5.4)$$

The hydrostatic pressure p_H , is given by;

$$p_H = -\left[\frac{\sigma_x + \sigma_y + \sigma_z}{3}\right] \quad (5.5)$$

where the directional stresses σ_x , σ_y , and σ_z are positive when tensile. Compressive stresses close cracks and act to reduce the shear stress criterion and increase life. Fatigue experiments performed under conditions of multi-axial stress states by Van Dang et al (1985) were used to develop this relation.

At high alternating stresses the fatigue threshold reduces. This physical behaviour is reflected in the life formulation, in terms of a fatigue shear stress limit, τ_u which is dependent on the maximum absolute value of the shear stress, $\tau_{\max}$. The relation and constants for bearing steels are;

$$\tau_u = 0.266 \text{ GPa} \quad \text{if } |\tau_{\max} + 0.3 p_H| \leq 0.6 \text{ GPa} \quad (5.6)$$

$$\tau_u = 0 \quad \text{if } |\tau_{\max} + 0.3 p_H| > 0.9 \text{ GPa} \quad (5.7)$$

$$\tau_u = 0.266 \left(1 - \frac{|\tau_{\max} + 0.3 p_H| - 0.6}{0.9 - 0.6}\right) \text{ GPa} \quad \text{if } 0.6 \text{ GPa} < |\tau_{\max} + 0.3 p_H| \leq 0.9 \text{ GPa} \quad (5.8)$$

Bearing fatigue life tests performed at SKF ERC have been used to verify the fatigue stress criterion and the fatigue shear stress limit.

In summary then, the procedure is, for each element in the life grid;

- (a) Perform the contact analysis between the roller and the indented surface for several roller positions. Calculate the subsurface stress field for each position¹⁶.
- (b) Add the residual stresses, from the FE analysis, to each life grid element. Determine the directional stresses, σ_x , σ_y , σ_z and hence the hydrostatic pressure for each element and for each overrolling position.
- (c) Determine the shear stress at each angle θ (say from 0° to 360° in steps of 10°) for each rolling position. Correct by the hydrostatic pressure and determine the maximum amplitude of the sum across the rolling positions (i.e. τ_a).
- (d) Determine the fatigue stress limit, τ_u from the maximum absolute value of the alternating shear stress (found at angle θ).
- (e) Calculate the value of the life relation, (τ_a/τ_u) for each element in the mesh. The relative life is then the summation of all these lives, divided by some reference life. A contour map of these life values, can be drawn to show the regions of high fatigue risk (see for example Dwyer-Joyce et al 1990).

5.2.3. Fatigue Life Parameters

In the following analysis various parameters are varied in the theoretical model; the effect of these parameters is assessed with reference to fatigue life. The size of the dent, relative to the size of the contact, and the slope of the dent are important in the prediction of fatigue life.

Lives are presented as reduction factors relative to the smooth contact life (where no indent is present). This has the advantage of separating the life reduction of the debris dent from that caused by increasing loads or contact widths.

Figure 5.4 shows the dimensions of the dent with respect to the contact. The dent slope is defined as $\tan^{-1}(\delta/\alpha)$. The dent size relative to the contact width is α/a .

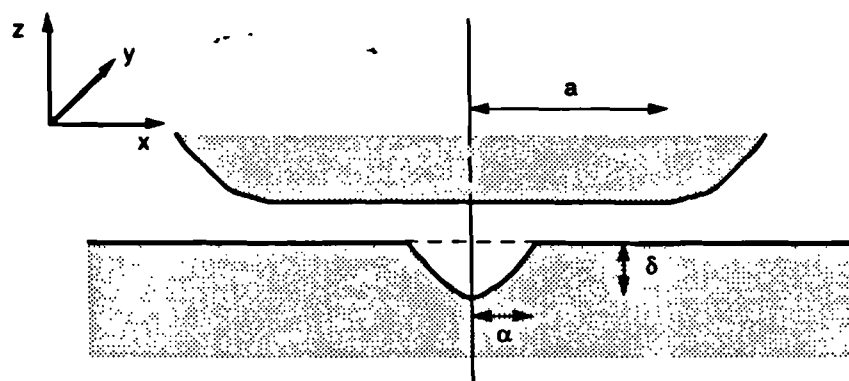


Figure 5.4. Parameters used in the theoretical predictions of fatigue life of indented contacts.

¹⁶ In some of the fatigue life calculations presented here only one ball/roller position was used, since this usually determines the worst case stresses (and so maximum amplitude of shear stress) and reduces computational time.

Since the analyses are axi-symmetric or plane strain their is only one contact dimension. The direction of rolling in Figure 5.4 is in the plane of the paper, so the contact width corresponds to the semi-minor axis of a ball bearing contact ellipse.

5.3. Point Contact and Axi-symmetric Dent

In this analysis the effects of a modelled axi-symmetric dent on fatigue life were studied. The tungsten carbide indenter model described in §4.4 was used to determine displaced shapes and residual stresses. The loads after an indentation of $a/R=0.26$ were used; this step was considered to be the highest loading where the residual stresses were still reasonably accurate (higher loads tended to introduce numerical errors as described in §4.3.4). The displaced shape and residual stress information was supplied to co-workers at SKF ERC who performed the fatigue life calculations.

The analysis performed at SKF ERC (and described in Beghini et al 1992) consisted of the determination of the contact pressure (after the method of Webster 1986) and the application of this pressure at a single location with relation to the dent (i.e. a quasi-dynamic overrolling analysis, as described in §5.2.1, was not performed). The influence on life caused by the overrolling process can be approximated by the single application of a contact pressure (as described by Lubrecht 1990).

Table 5.1 shows the reduction in lives caused by a single dent for three separate contact conditions. The lives are compared with the smooth case; thus the effect of the increase in size of stressed volume, caused by the higher pressure, is removed. The reduction in life caused by dent shape and residual stresses are compared¹⁷. In all cases the residual stresses have a lesser influence than the dent shape.

Pressure, GPa	Hertz width, μm	Life reduction factors (thousands)		
		Dent alone	Residual stresses	Dent and residuals
3.0	300	1.3	0.6	1.0
2.5	250	21	4	13
2.0	200	1800	150	1100

Table 5.1. Life reduction factors for a surface dent considering the effects of dent shape and residual stresses for three contact conditions.

For lightly loaded contacts the peak shear stresses are nearer the surface; when an indent is present these interact with the near-surface residual stresses and have a greater influence on fatigue life. The dent also occupies a greater proportion on the contact area; and thus the high contact stresses take a larger proportion of the stressed volume. A surface indentation therefore has a significant effect on lightly loaded applications. At higher loads the contact stressed volume is greater and the part played by the high dent stresses is reduced. Thus the fatigue life reduction factor is less.

Since the residual stress effect, as shown in Table 5.1, was small compared to the dent geometry effect the following analyses were performed for a dent shape only.

¹⁷ It should be noted that the life reduction factors are not additive (i.e. the dent factor plus the residual factor does not equal the combined factor). This is because the residual stresses act independently of the dent overrolling stresses; an element may have a high risk below a dent shoulder but this may be reduced by compressive residual stresses in that region. The total fatigue life reduction is the sum of all of these elemental risks.

The effect of the size of the dent is shown in Figure 5.5. The life reduction factor is plotted against the ratio of dent half width to contact half width.

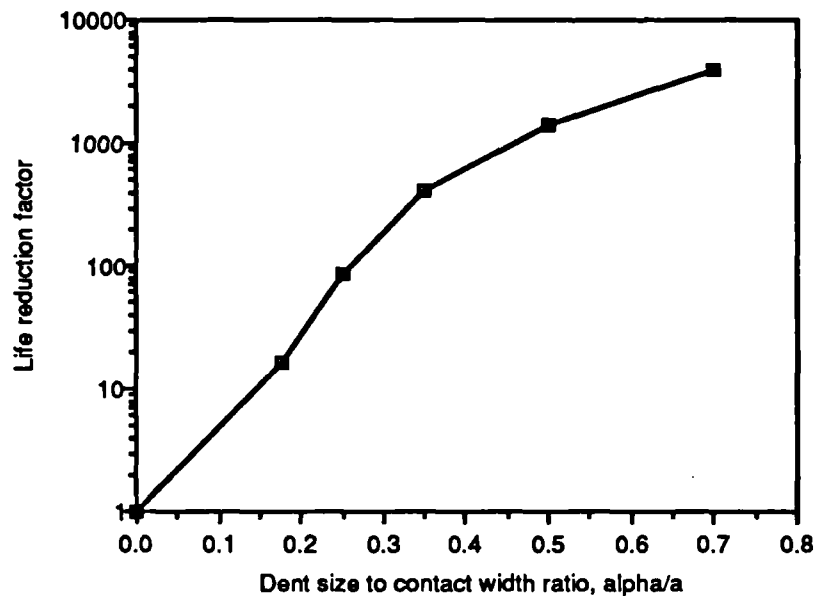


Figure 5.5. Life reduction factor for dents of various size with respect to the contact width. The dent slope is constant at 6.8° as is the maximum contact pressure at 3.0GPa.

A large dent causes high contact pressures and therefore subsurface shear; these will represent a greater proportion of the overrolling stresses and therefore greatly reduce life. This also shows the importance of bearing size. A small bearing, with a small contact area, will be more sensitive to surface damage than a large bearing. Another important feature is the slope of the dent. Figure 5.6 shows a plot of life reduction against dent slope.

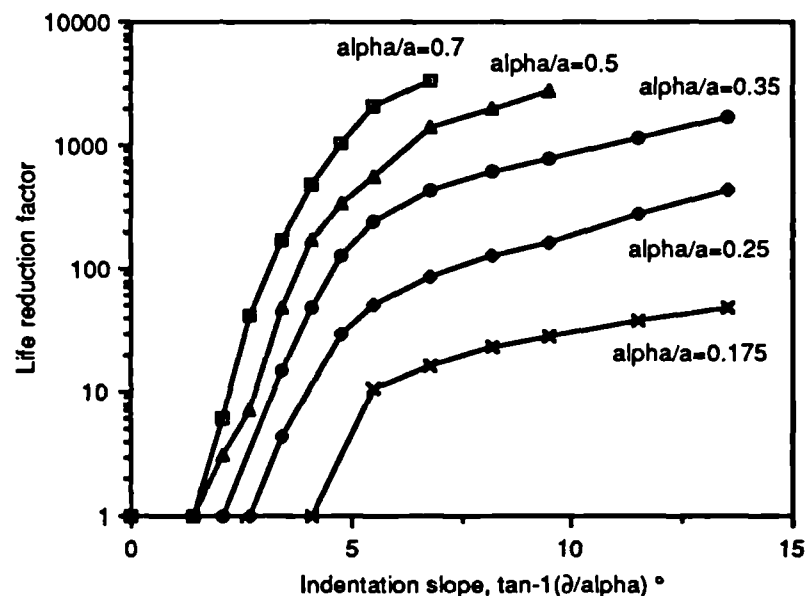


Figure 5.6. Plot of life reduction (compared with smooth case) against slope of surface dent (slope $=\tan^{-1}d/\alpha$) for five different contact patch sizes (α/a) 0.175, 0.25, 0.35, 0.5, and 0.7.

The slopes were obtained by the appropriate scaling of the dent predicted from the finite element analysis. The data is repeated for five different dent sizes (α/a) whilst the contact pressure is maintained at 3GPa. At low dent slopes the contact elastically conforms around the dent and fatigue life is little affected. Steep sloped dents induce high contact pressures and subsurface shear stress; life is therefore greatly reduced. Once again the larger dents cause a greater fatigue life reduction. The worst combination, therefore, is a large steep sloped indentation on a small bearing surface.

5.4. Line Contact Analysis and Asperity Indentation

A line contact indentation was also carried out using the same method. In this instance plane strain analysis was used. The indentations were more representative of that caused by a surface ripple or grinding ridge transverse to the direction of overrolling.

A typical finite element mesh used is shown in Figure 5.7. The two bodies were mirror-symmetric made up of 250 eight noded reduced integration plane strain elements (CPE8R); the contact was defined using six noded interface elements (INTER3). The lower body contained a bump of 100 μ m half-width; five meshes were created with bumps of slope 2.5°, 5°, 7.5°, 10°, and 15°. The upper body surface was curved. Both bodies were modelled as elastic-perfectly plastic and the effects of friction were neglected.

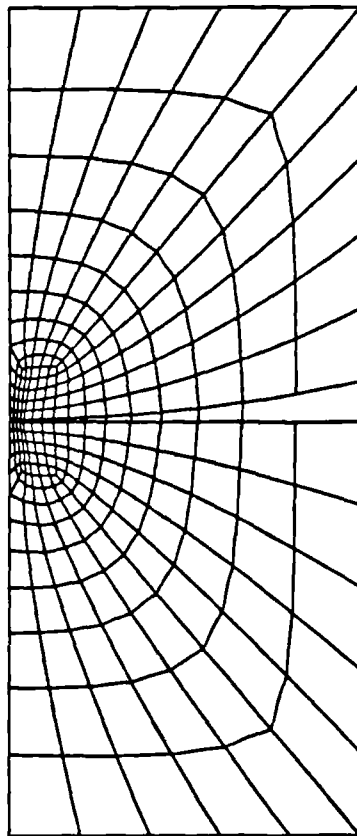


Figure 5.7. Typical finite element mesh used for the contact fatigue life study of plane strain indentations.

Two distinct indentation processes were applied. The upper body was displaced onto the lower until the bump was just flattened, the bulk surfaces were only just touching and a surrounding Hertzian pressure did not develop. The second loading case consisted of the displacement of the

upper body further onto the lower until an elastic contact developed around the bump (the displacement was chosen such that a peak Hertzian contact pressure of 3GPa would have been generated if no bump were present). Figure 5.8 shows the contact pressures generated for these two loading conditions.

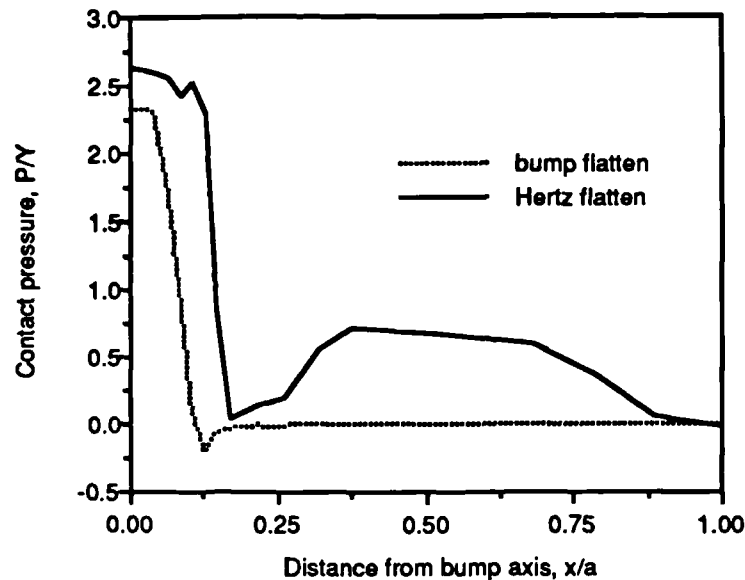


Figure 5.8. Contact pressures generated by the loading of a 10° bump against a curved surface. Two cases; flattening of the bump, and application of a surrounding Hertzian pressure.

Unloading the upper body, by displacing it away from the lower, gives residual stresses and displaced shapes. Figure 5.9 shows the residual shapes left by the indentation by the five different slope bumps.

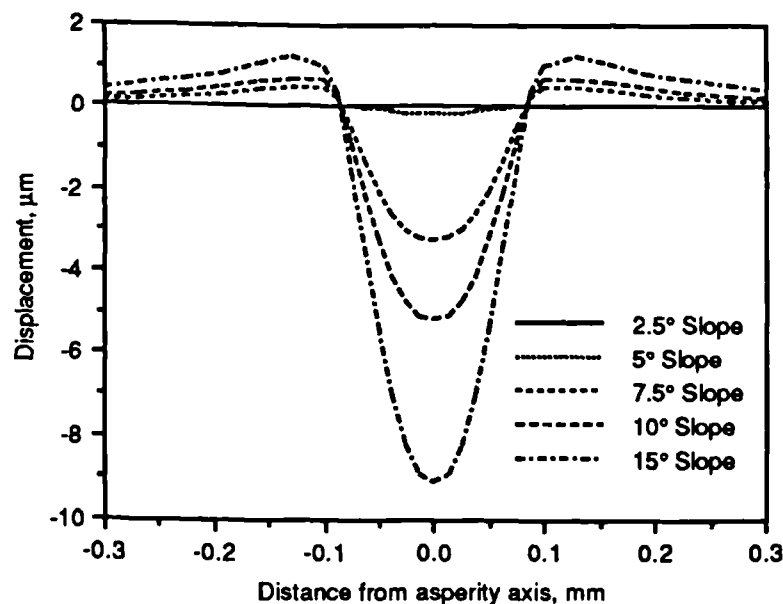


Figure 5.9. Residual dent shapes left by the indentation by the five different slope bumps (after full loading in Hertz contact).

The slopes of the resulting indents are considerably less than the slopes of the bumps. Table 5.2 compares the dent geometry with the indenting bump geometry, pressed in the surrounding Hertzian contact.

Indenter slope	Dent slope $\tan^{-1}(\delta/\alpha)$	Residual dent depth, μm	Loaded dent depth, μm
2.50	0.10	0.18	4.47
5	0.87	1.52	8.75
7.5	1.86	3.25	13.2
10	2.98	5.28	17.6
15	5.20	9.10	26.8

Table 5.2. Comparison of indenter slopes with slopes of final dent. All indentations performed in a surrounding contact (i.e. Hertzian flattening).

The same fatigue life analysis was carried out on each of these five cases. Table 5.3 shows how the dent shape and residual stresses effect the fatigue life for a 15° bump.

Case	Life Reduction Factor caused by:		
	Dent alone	Residuals alone	Dent and residuals
Bump flatten	27	1.4	31
Hertzian flatten	1920	13.9	851

Table 5.3. Life reduction factor caused by the dent shape and residual stresses from the flattening of a 15° bump in a simple and Hertzian contact (overrolling contact conditions $p_0=3.0\text{GPa}$, $b=0.5\text{mm}$).

Again the dent shape appears to be the most important factor in determining the fatigue life reduction. The residual stresses have a slightly beneficial effect; they are largely compressive and will, for many of the life grid elements, reduce the risk term. The inclusion of the bulk pressure in the flattening of the dent has a large effect on fatigue life. In the Hertzian flattening case a deeper dent was generated and residual stresses were higher. This was thought to be the more representative case; the rest of analysis was performed using this model. Table 5.4 shows how the fatigue life is affected by the slope of the indenting bump.

Indenter Slope	Life Reduction Factor caused by:		
	Dent alone	Residuals alone	Dent and residuals
2.5°	0.96	1.0	0.98
5.0°	0.78	1.5	0.94
7.5°	0.61	1.5	0.88
10.0°	38	1.9	46
15.0	1920	13.9	850

Table 5.4. Life reduction factor caused by the dent shape and residual stresses from the flattening of a five different slope bumps in a Hertzian contact (overrolling contact conditions $p_0=3.0\text{GPa}$, $b=0.5\text{mm}$).

It is clear that, as with the previous axi-symmetric model, the slope of the indenter has a large effect on life reduction. High sloped indents lead to high overrolling contact pressures and sub-surface shear. This data is replotted as the life reduction factor against dent slope (rather than indenting bump slope) in Figure 5.10.

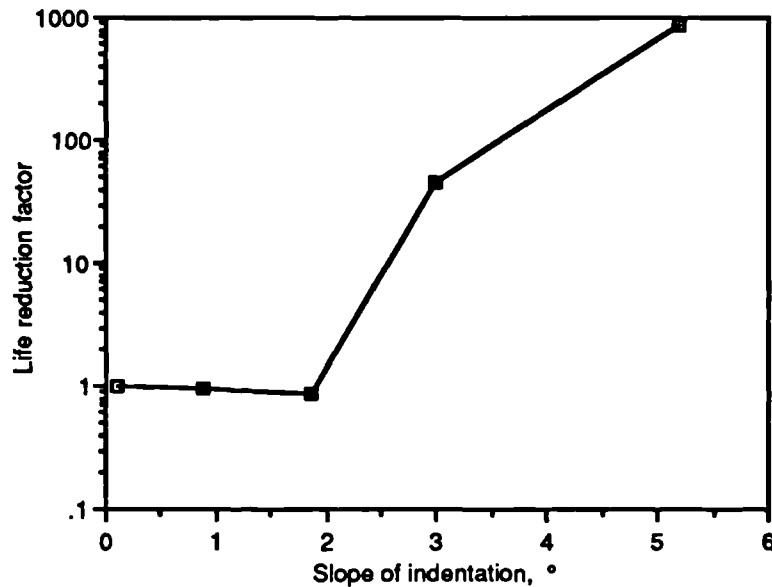


Figure 5.10. Plot of life reduction factor against slope of the surface dent, $\tan^{-1}(\delta/\alpha)$. The relative size of the dent remains constant ($\alpha/a=0.2$) as does the overrolling contact pressure ($p_0=3.0\text{GPa}$).

At the low indentation slopes, less than 2° , a small increase in life is noted. This is caused by the increased conformity of the roller over the dent which in turn leads to lower contact pressures. When a full overrolling analysis is performed (rather than using a single loading position to obtain sub-surface stresses) this effect disappears since a roller positioned over the shoulder of the dent will not show this same conformity (see Lubrecht 1990 for a full investigation).

This data compares well with the axi-symmetric analysis displayed in Figure 5.6 (using the $\alpha/a=0.25$ curve as the closest dent size comparator). Both show a dent slope threshold, for no reduction in fatigue life, of about 2° . Life reductions for the planar case are somewhat higher than those calculated for the axi-symmetric case; a planar surface groove presents a more serious case than a circular dent.

The graph presented as Figure 5.11 shows how the life reduction factor varies with the dent width to contact width ratio (α/a). As the contact width decreases, relative to the dent, the life reduction becomes more severe. The dent and its local stress raising effects are large compared with the contact, and therefore play a greater role in determining the fatigue life.

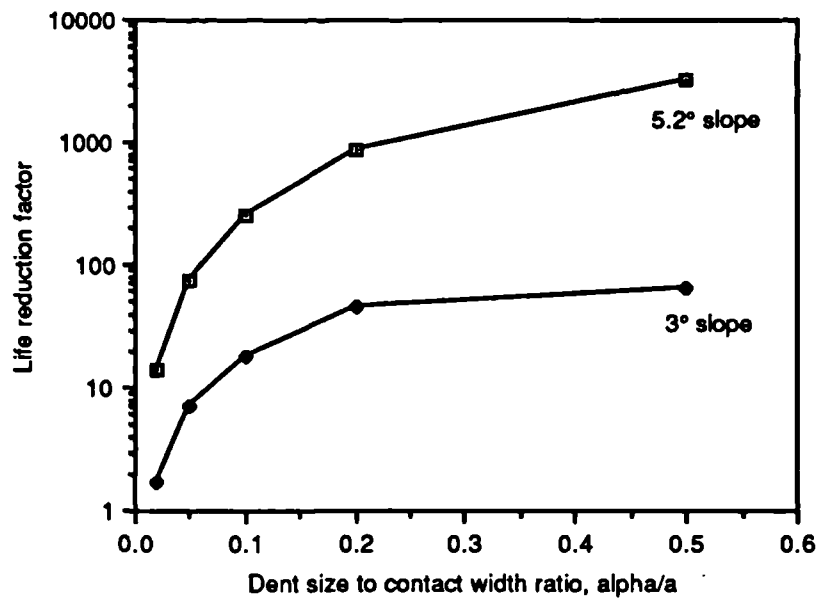


Figure 5.11. Influence of dents of slope 3° and 5.2° (created from indentors of slope 10° and 15°) for different ratios of dent width to contact width (α/a). In both cases the dent width remains constant at 0.1mm, as is the peak contact pressure at 3.0GPa.

Figure 5.12 shows the effect of changing the overrolling contact pressure on fatigue life reduction factor.

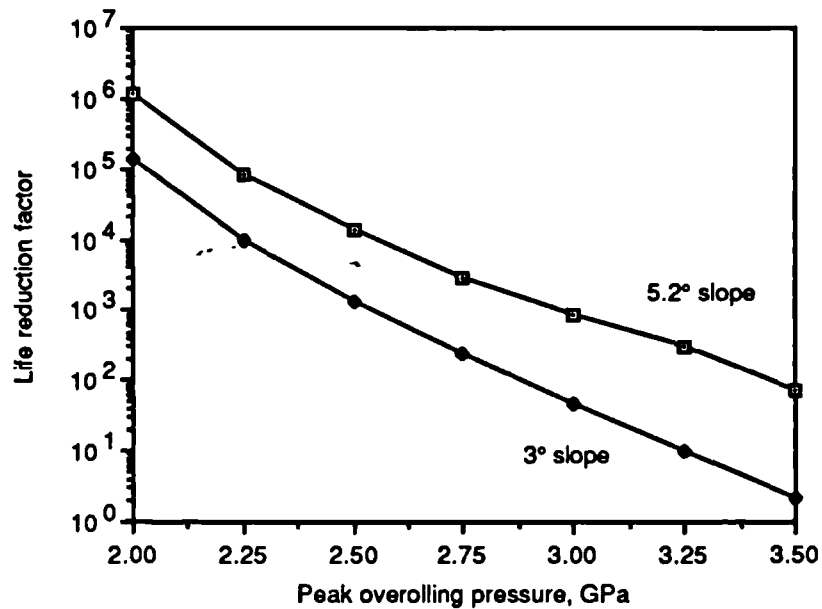


Figure 5.12. Influence of dent of slope 3° and 5.2° (created from indentors of slope 10° and 15°) for different contact pressures. In both cases the dent width to contact width ratio remains constant at 0.2.

As the contact pressure increases the effect of the dent on life¹⁸ is reduced. The dent is a localised effect and independent from the bulk of the overrolling pressure; at higher contact pressures the localised dent pressures play a proportionately smaller part in the creation of the sub-surface stressed volume. Its effect on fatigue life is therefore reduced.

5.5. Asperity Overrolling and Fatigue Life

This study was aimed at predicting the effects of the overrolling of planar asperities on the fatigue life of gear tooth contacts. Micropitting frequently occurs on gear tooth faces. Small pits, appearing as a surface crazing, are generated and can propagate to fatigue spalls. The formation of these fatigue pits appears to be strongly dependent on stress state, surface finish, sliding, and lubrication. The purpose of this study was to model the overrolling of a surface containing an asperity or ridge with and without friction and sliding to determine the effect on fatigue life.

A typical finite element mesh is shown in Figure 5.13. The two bodies are modelled using eight noded plane strain quadrilateral elements (CPE8R) with a contact of six noded interface elements (INTER3). A mirror symmetric model was unsuitable since sliding and friction were introduced. The lower body contains a surface ridge; five different meshes were generated with ridges of slope 1°, 2.5°, 5°, or 10°. The base of the lower body was simply supported whilst the upper body was displaced, along a forty five degree vector, onto the lower until the ridge was flattened as far as the bulk surface level.

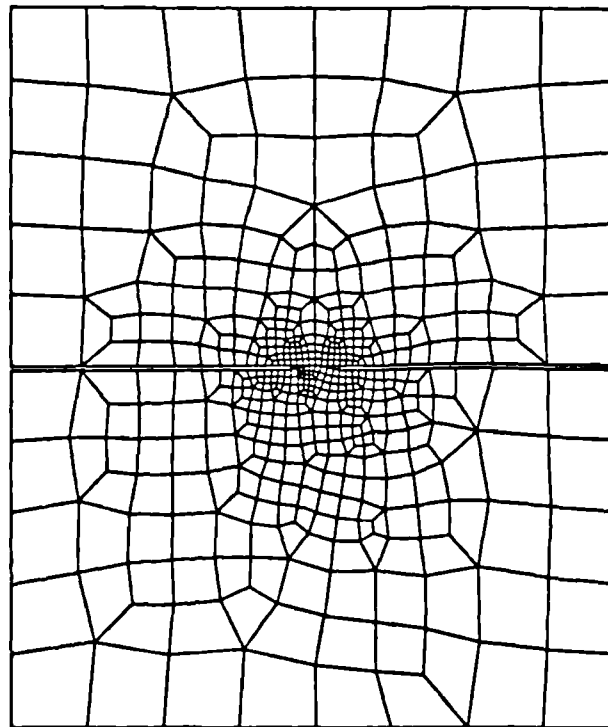


Figure 5.13. Finite element mesh used in the study of asperity flattening. The lower surface contains a ridge (ridges of slope 1°, 2.5°, 5°, and 10° were used).

¹⁸ The life itself is very dependent on contact pressure, since the stressed volume is increased (e.g. life at a pressure of 2GPa is 600,000 times greater than that at 3GPa). All the data here is presented as life reduction factors; and thus this stressed volume effect is removed.

Friction coefficients of 0.05 or 0.1 were applied during the flattening process. Again residual stresses and displaced shapes were obtained from the finite element analysis. These were used in the fatigue life model to predict life reduction factors (compared with the life for a smooth case).

Table 5.5 shows the life reduction factors for the three different asperity slopes being flattened and overrolled with a friction coefficient of 0.1. As in the previous studies life is highly dependent on slope. Steep asperities induce high overrolling stresses and, when friction is present, high shear stresses which reduce life dramatically.

Slope	Life reduction factor
1°	1.3
2.5°	6500
5°	27000

Table 5.5. The variation of life reduction factor with the slope of the surface asperity. Friction coefficient of 0.1 residual stresses included.

The high contact pressures, predicted from the elastic contact model, were thought to be unrealistic. These contact pressures were, therefore, truncated. The threshold was chosen as a typical peak contact pressure for a shakedown gear tooth surface (i.e. all the surface irregularities have been flattened by repeated overrolling). This pressure was determined by measuring worn gear tooth surfaces and using the profiles in the elastic contact model (§5.7 discusses this approximation further).

Table 5.6 shows the effect of the friction coefficient on life for a 5° degree asperity, including residual stresses. The increase in friction causes the transfer of high shear stresses during overrolling.

Friction coefficient	Life reduction factor
0	19
0.05	2400
0.1	27000

Table 5.6. The variation of life reduction factor with the overrolling friction coefficient for an asperity of slope 5° with residual stresses.

The plots included as Figures 5.14, 5.15, 5.16, 5.17, and 5.18 are fatigue life 'risk maps'. They are plots of the fatigue life term $(\tau_a - \tau_u)^c$ from equation 5.3, and therefore present a picture of the likelihood of a fatigue crack initiating from a particular subsurface region. The life term is plotted as a logarithmic scale so the difference between the top and bottom is several orders of magnitude. In each case the plot is a section parallel to the direction of rolling (as in the lower body of Figure 5.3) and the asperity, when present, is located on the upper face in the centre.

Figure 5.14, shows the risk map for a smooth surface. The regions of high risk are in a band sub-surface; this is caused by the peak Hertzian contact shear stress which is located sub-surface. The maximum amplitude of shear stress as the roller passes over is therefore a sub-surface band. The wavy shape of this band is caused by the discrete number of overrolling solutions; using more overrolling solutions would give a band bounded by a straighter line.

Figure 5.15, shows the effect of an asperity located at the surface, the residual stresses are not considered. The overrolling has resulted in high contact pressures in the regions of the asperity shoulders which lead to high shear stresses and therefore a high risk. The hydrostatic pressure below the surface under the asperity has tended to reduce the effect of the sub-surface band. Figure 5.16 shows the same plot but with the inclusion of residual stresses. These are mostly compressive and have therefore reduced the fatigue life criteria in most places, high tensile stresses exist and lead to high risk regions around the asperity shoulders.

Figure 5.18, shows the effect of including friction, with a sliding direction from left to right. High shear stresses, in the direction left to right, have been induced by the overrolling; these reinforce those from the indent on the left side and reduce those on the right. Hence, an asymmetric risk pattern is produced with the regions of highest risk at the trailing edge of the contact. Indeed, this is the mechanism observed with most surface initiated fatigue failures, a fatigue crack initiates from the trailing edge of a dent or surface ridge (the photograph included as Figure 5.1 shows the fatigue spall initiated from this region). The inclusion of the residual stresses in the analysis, Figure 5.18, reduces the life risk; as before the high compressive residual stresses reduce the life term.

The overall conclusion from this study was that surface asperities reduce life, the steeper the slope, the more dramatic the reduction. Sliding induces high shear stresses at the trailing edge of the dent and leads to a high risk, and likely crack initiation, in this region. In terms of gear tooth micro-pitting this would imply that pitting failure will occur on rough surfaces at regions remote from the pitch line.

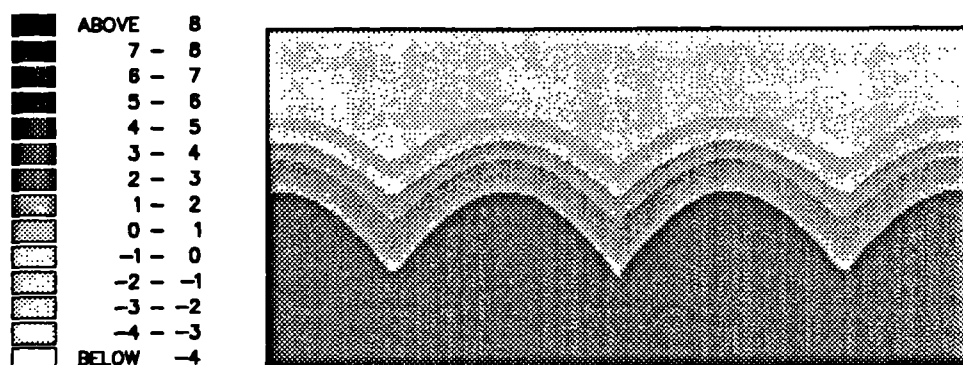


Figure 5.14. Fatigue life risk map for a smooth surface, no friction.



Figure 5.15. Fatigue life risk map for a surface containing a 10 slope degree asperity, residual stresses not included, no friction.

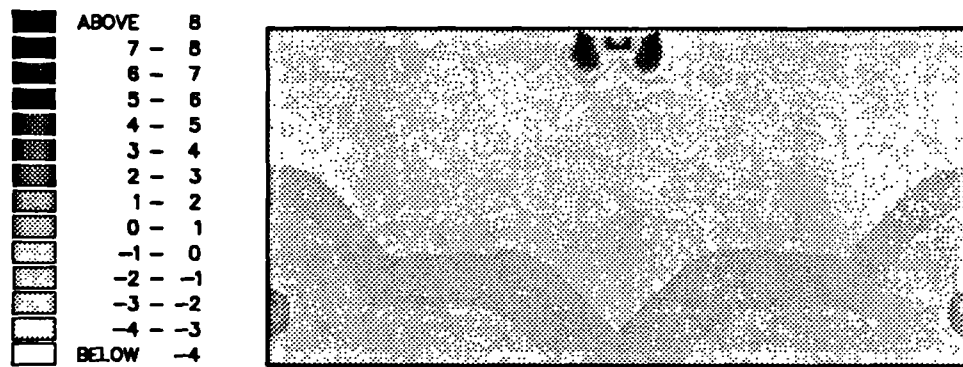


Figure 5.16. Fatigue life risk map for a surface containing a 10 slope degree asperity, residual stresses included, no friction.



Figure 5.17. Fatigue life risk map for a surface containing a 10 slope degree asperity, residual stresses not included, friction coefficient 0.1.



Figure 5.18. Fatigue life risk map for a surface containing a 10 slope degree asperity, residual stresses included, friction coefficient 0.1.

5.6. Experimental Data

Lorosch (1983) tested bearings with surface damage caused by indentation with a tungsten carbide ball. The tests closely resemble the analysis described in §4.4. Table 5.7 is a summary of his results. As before, the results have been normalised using the dent slope and the ratio of dent size to contact size (the original experiments were performed on three different sized and geometry bearings with contact semi-widths¹⁹ of 0.08mm, 0.4mm, and 0.43mm)

¹⁹ Lorosch only gives the length of the contact ellipse, b , from this a has been estimated.

Indent slope, $\tan^{-1}(\delta/\alpha)$	Dent width/contact width, α/a	Life reduction factor
14.5	0.63	58
31	1.25	196
31	1.88	400
14.9	1.88	100
15.6	3.13	175
4.5	3.13	6.3
14.5	0.13	1
31	0.38	26
14.5	0.12	1.25
31	0.35	8.33

Table 5.7. Lorosch's (1983) results for the fatigue life reduction caused by indentations of various slope under different contact conditions.

Increasing the slope and size of the indent causes a reduction in fatigue life. These features were noted in the theoretical predictions. Although the theoretical predictions show much greater life reduction factors than the experiments. For example dents of slope 14.5° with a size ratio of 0.12 show a life reduction factor of 1.25; the theory predicts a factor of about 30 for similar conditions.

Figure 5.19 shows a plot of the life reduction factor against the product of the indentation slope, $\tan^{-1}(\delta/\alpha)$ and size to contact width ratio, (α/a) .

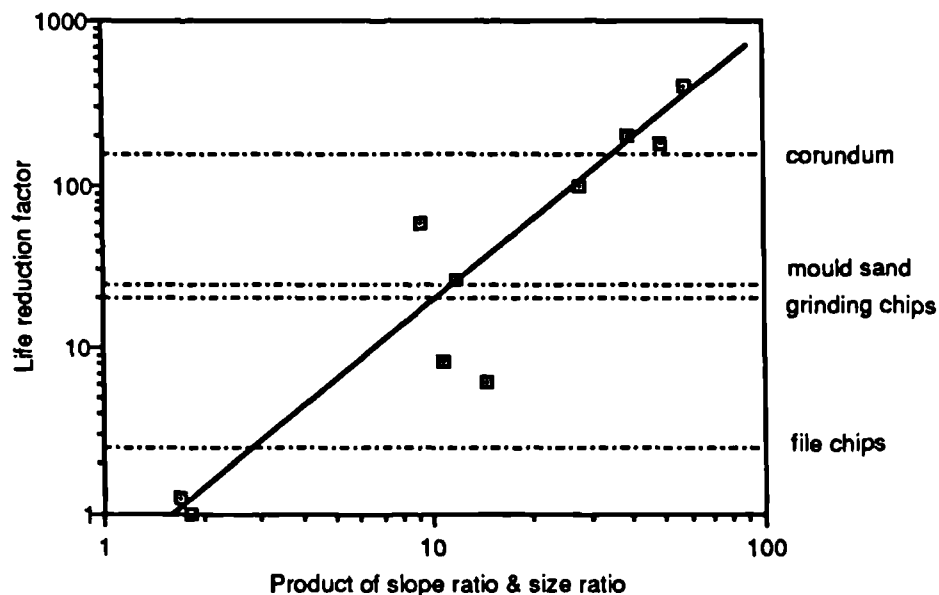


Figure 5.19. Fatigue life reduction factor against the product of indentation slope, $\tan^{-1}(\delta/\alpha)$ and size to contact width ratio (α/a) . Over plotted are the results of bearing fatigue tests with lubricants contaminated with various debris materials. Extracted from the data of Lorosch (1983).

This shows how the large steep sloped dents have the greatest effect on fatigue life. Lorosch also performed some experiments using debris particles in the lubricant (rather than pre-indented raceways). The fatigue life reduction factors are plotted on the figure for comparison; the data demonstrates where the real debris dents stand in relation to the artificial indents.

5.7. Comments on the Method

Several assumptions are inherent in the modelling process. This section describes these limitations and discusses their possible effect on the results.

- (a) The finite element process models a single indentation event. In reality an indent is overrolled many times and would *shakedown* to some new profile. Shoulders will be less steep and residual stresses will build up. The elastic contact model will thus predict lower pressures and sub-surfaces stresses causing an increased life. Whereas a build up of residual stresses may lead to a decreased life.
- (b) The elastic contact model does not model plastic flow and plastic contact pressures. The contact pressures predicted over the shoulders of the deep dent were above yield. The stresses generated at these points are therefore unrealistically high and the effect on life too severe. One approach (which was used in the latter study described in §5.5) is to limit these elastic pressures at some 'shakedown' value. The difficulty lies in choosing this value, the result are highly dependent on this choice. Another, and more rigorous, approach would be to use a shakedown dent in the analysis (i.e. model plastic shakedown at the finite element stage). The contact would then again be elastic since all plastic deformation would have occurred in early overrollings. The fatigue calculation would then be based on a justifiably elastic assumption.
- (c) The contact model assumes a dry contact; no attempt is made to model the separating lubricant film. Work carried out using a numerical elastohydrodynamic contact in conjunction with the contact fatigue life model (Lubrecht 1990) showed that the dry contact assumption differs from the lubricated analysis by only 6%. Lubrication tends to moderate contact pressures and reduce sub-surface stresses.
- (d) The finite element model of indentation is a poor approximation to the particle overrolling process. Ceramic particles become fully imbedded into a surface whilst ductile particles are flattened out. Both of these processes involve gross plastic flow and are difficult to model using the finite element method (the mesh distorts considerably and requires continual remeshing operations).

5.8. Conclusions

Finite element analysis provides a flexible method for the modelling of surface indentation. The results from the unloading step of these finite element studies were used in conjunction with a rolling contact fatigue model to predict the life reduction of a contact with an indented surface. In conjunction with co-workers at SKF ERC, three separate analyses were performed, an axis-symmetric spherical indenter, a planar indentation by a ridge, and the tractive overrolling of a planar asperity.

The analyses showed similar features, the general conclusions are as follows;

- (a) Life reduction is strongly dependent on the shape of the indentation. Steep sloped dents produced high overrolling pressures and sub-surface shear.

- (b) The effects of the residual stresses from an indentation have only a minor effect on fatigue life; and in some cases this effect is beneficial.
- (c) The size of the contact with respect to the dent size is important. If the local dent stresses dominate the overrolling stresses, then life is greatly reduced. In addition at low loads, and hence contact width, the near surface residual stresses from the dent interact with the sub-surface overrolling stresses; this causes major life reduction. Therefore, deep, steep-sloped dents on small lightly loaded bearings will greatly reduce life; whilst a larger bearing may accommodate a greater level of damage.
- (d) Dents of slope less than about 2° had little effect on fatigue life.
- (e) The mechanism for fatigue crack initiation at the trailing edge of a dent has been explained. Shear stresses from the overrolling are superposed on those from the indentation. At the leading edge of the dent these act in opposition, whilst at the trailing edge they reinforce; higher shear exists in this region and fatigue cracking is promoted.
- (f) The results of the axi-symmetric indenter case were compared with the experimental results of other workers. Similar trends were noted although magnitudes differed somewhat.

This method of finite element indentation and contact fatigue modelling has several limitations. However, the study has been instructive and has shown the mechanisms of plastic indentation, the nature of residual stresses, and the important parameters relevant to fatigue life.

CHAPTER 6

ABRASIVE WEAR SURVEY

6.1. Introduction

This chapter has been set aside for a review of the published research on abrasive wear. The survey begins with a brief discussion of wear processes and a classification of abrasive wear. The abrasive wear of rolling bearings by lubricant contaminants is a special case of abrasive wear; little work has been done in the field. For background information, therefore, it has been necessary to study analogous abrasive wear mechanisms. Useful information has been found in studies on two-body, abrasive paper or grinding wheel type, wear.

6.2. Wear

The failure of machine parts generally occurs by excessive deformation, fatigue, or wear. The first two of these mechanisms are fairly well understood and quantitative failure laws are in common use. Wear, however, is far from a known science; it is frequently approached empirically or from a qualitative point of view.

The Institution of Mechanical Engineers has defined wear as the;

"progressive loss of substance from the surface of a body brought about by mechanical action (usually it reduces the serviceability of a body, but it can be beneficial in its early stages in running-in)."

This process of removal of substance has been described in terms of several contrasting mechanisms;

- (i) Adhesive wear. Material loss is caused by asperity interaction between two poorly lubricated sliding surfaces.
- (ii) Chemical wear. Material loss by chemical reaction, such as corrosion reactions.
- (iii) Fretting wear. Low amplitude oscillatory motion in combination with corrosion cause oxide formation and abrasion by trapped particles.

- (iv) Abrasive wear. A hard rough surface slides against a softer surface, digs in and, ploughs out grooves. Alternatively, particles may become introduced between sliding surfaces and plough out grooves of material.

It is the last of these mechanisms that is of interest to those studying the effects of lubricant contamination on rolling or sliding contacts. It has been said (Eyre 1976) that some 50% of wear occurrences in industrial applications can be attributed to abrasive wear.

6.3. Archard's Wear Equation

One of the more established wear laws was developed by Archard (1953). He stated that the wear volume, V was proportional to the applied load, W , the sliding distance, l and inversely proportional to the hardness, H .

$$V = \frac{kWl}{3H} \quad (6.1)$$

The constant, k is known as the wear coefficient. So given a particular application the coefficient, k can be used as a comparator to determine the wear resistance of different materials.

The model was first developed for adhesive wear but has also been applied to other wear modes. Rabinowitz (1965) adapted the form of the equation for abrasive wear; he considered a conical asperity (or abrasive grain) ploughing a surface (see Figure 6.1).

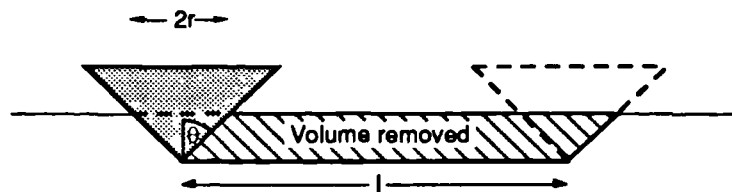


Figure 6.1. Rabinowitz's simple model for abrasive wear.

The penetration into the surface is controlled by the hardness of the material and the load, if the indentation is fully plastic;

$$W = H \pi r^2 \quad (6.2)$$

The volume of material removed (shaded in the diagram) is given by;

$$V = r^2 \cot \theta l \quad (6.3)$$

The total volume removed is the result of many such individual contacts; based on an average value of the cone angle, θ . The total volume lost is obtained by combining equations (6.2) and (6.3) to give;

$$V = \frac{\cot \theta W l}{\pi H} \quad (6.4)$$

This is equivalent to the Archard equation, replacing $\cot \theta / \pi$ by $k/3$. Rabinowitz (1981) has compiled a large base of typical wear coefficient data under adhesive, abrasive, corrosive and

fretting wear (see Figure 6.2). The data is presented as order of magnitude quantifiers and not as an accurate design guide; nevertheless, this sort of information is of real practical use to the design engineer.

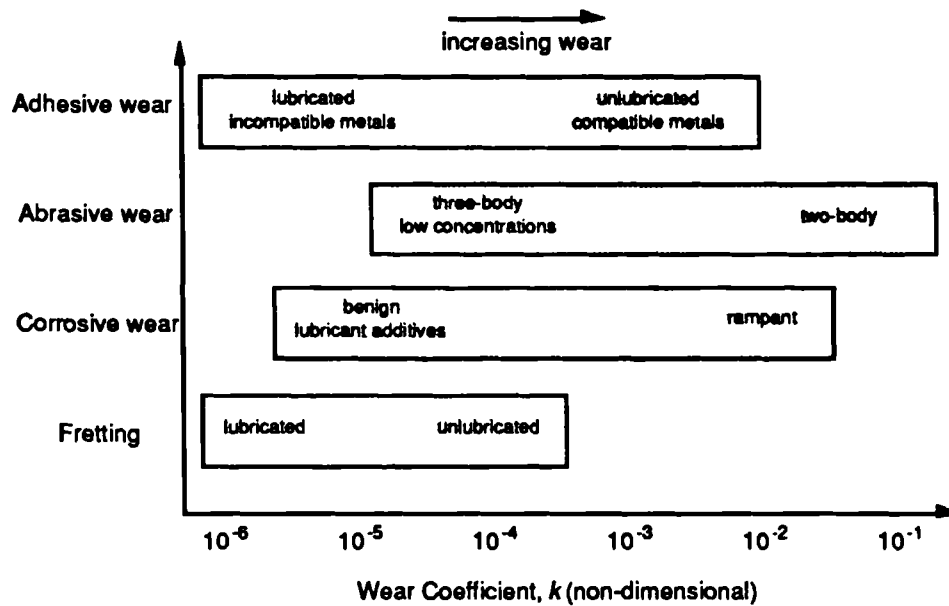


Figure 6.2. Typical wear coefficients for various sliding wear mechanisms.

In general the mechanism of three-body wear falls somewhere mid-range in this scheme of abrasivity. In §7.4.5 an attempt is made to generate wear coefficients from the experimental data in this study.

The collection of wear information into a database is a useful approach when the nature of the science is not well understood. In recent years, with the advent of fuzzy logic models, unstructured wear data from related systems can be interpreted for directly useful information (see for example Stupka and Dohnal 1992).

6.4. Classification of Abrasive Wear

Burwell (1957) divided abrasive wear into two categories; 'two-body' and 'three-body' abrasive wear. Two-body abrasive wear occurs when a pair of opposing surfaces interact abrasively. Three-body abrasive wear occurs when abrading particles are interposed between two surfaces; the particles may move relative to one another and possibly rotate while sliding; material may be removed from either surface.

Later a further category was added, 'two and a half body', where the abraded particles are fixed to one surface but that surface is flexible and allows some accommodation of the opposing surface shape. Erosive wear is difficult to classify under either of the groups; in this mechanism fluid-borne abrasives impinge directly on a surface.

The two-body mode of abrasive wear is most commonly found in material removal operations, such as grinding and tool cutting, or when a rough surface wears a smoother surface by the action of cutting asperities. The two and a half body abrasive wear is found in emery paper abrasion operations.

The three-body wear mechanism can be further sub-divided into two categories; 'open' and 'closed' three-body wear. Open three-body abrasive wear occurs when the two opposing surfaces are far apart or when only one surface is involved in the wear process. Closed three-body abrasive wear occurs when the abrading particles are trapped between two opposing sliding or rolling surfaces; such as wear in contacts under conditions of particulate contaminated lubrication.

Misra and Finnie (1970) still further sub-divided the mechanism of three-body abrasive to include 'high-stress', and 'low-stress' variants. High-stress abrasion occurs when the abrasive particle is crushed during the wear the process, such as occurs in ball mill grinding. Whilst, in low stress abrasion the abrasive particles are not broken down during the wear process.

This rather complex categorisation of abrasive wear is displayed schematically in Figure 6.3.

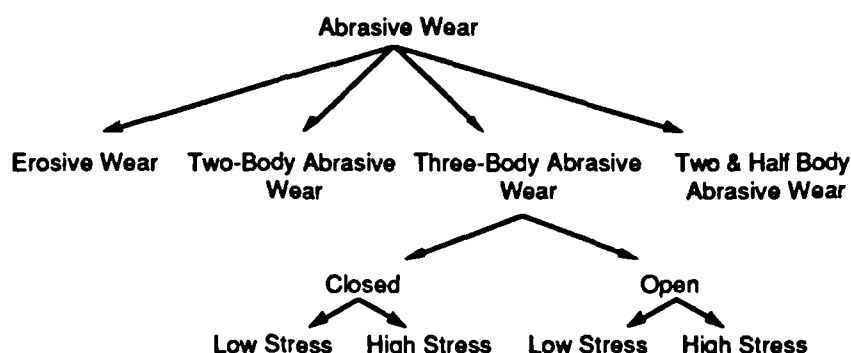


Figure 6.3. The classification of abrasive wear

It is the mechanism of closed three-body abrasive wear that is the most applicable to this research study, where debris particles are overrolled in a rolling element bearing. However, much of the wear process is common to the other abrasion mechanisms; for instance, the generation of a groove, ploughing, and indentation processes. It is therefore instructive to review abrasive wear under both two and three-body categories.

6.5. Two-Body Abrasive Wear

Research into the mechanisms of abrasive wear largely falls into two camps. The first approach has been based on experiments with abrasive papers, and using models based on ploughed material and the number of grits cutting. The other approach has been through experimentation with cutting tool type indentors; used to represent a hard asperity ploughing into a softer surface. The modelling in the later case has been using the slip line field method to predict regimes of cutting and plastic flow.

An early contribution to the understanding of two-body wear was made by Khrushchov (1957). Using pin on abrasive disk experiments the author developed a simple proportional relationship between resistance to abrasive wear and metal hardness. Heat treated steels were found not to follow the same relationship, with wear resistance found to be less than expected for a given hardness. The author explained that the wear resistance was independent of heat treatment; since the abrasion cutting process involves considerable work hardening. Thus the resistance to abrasion depends on the strength of the material in its maximum work hardened state.

Goddard and Wilman (1961) and later Stroud and Wilman (1962) further investigated wear by abrasive papers. They studied the grooves formed during the abrasive wear process. They noticed lateral pile-up of material and made estimates for the proportion of material removed

from the groove. They developed expressions for the ploughing and adhesion friction coefficients, from the material hardness and grit geometry. These were then used to predict the wear mass from the grit geometry and the supported loads. The model included factors for material pile-up; they also noted that not all contacts may be causing wear and included fractions of load supported by abrasive, metal, and oxide contact.

Sedriks and Mulhearn (1963) and Mulhearn and Samuels (1962) worked with both abrasive papers and model cutting tools. They realised that the cutting angle was important for abrasion and determined critical cutting angles for pyramidal sliders; these were then related to the cross-sectional area of the groove. They then measured the cutting angles present in sections of silicon carbide abrasive paper, and estimated the percentage of contacting grits which cut as between 7 and 10%. Further studies into the deterioration of abrasive papers were carried out. All this information was incorporated into a comprehensive model; the equation for wear mass, m_n for the n th abrasive grit pass was of the form

$$m_n = \frac{\rho f_n l W}{H k \phi} \quad (6.5)$$

where, ρ is the density, l the sliding distance, W the load, H the hardness, ϕ is a form factor for the shape of the groove, and f_n is the proportion of grits cutting (which deteriorated exponentially with time). The equation shows some similarities with the Archard wear equation (6.1) with factors relevant to the abrasive paper wear process. An extensive scheme of well designed experiments were performed to validate these factors.

Experiments on the effect of the size of the abrasive grit were performed by Nathan and Jones (1966). Bonded abrasive particles sized from 35 to 710 μm were rubbed against various metal surfaces. A critical size of particle (around 100 μm) was found to exist, above which wear was greatly reduced. Tests on the effect of velocity, load, and sliding distance were found to be in agreement with the Archard wear equation.

The relative hardness of abrasive and substrate were investigated by Richardson (1967 & 1968). Again using abrasives bonded to a backing paper and with a wide selection of materials. He pointed out that the wear resistance of a material is specific to the abrasive (to within 10-20%) and not an inherent property (in contrast to the postulations of Khrushov). The effects of material microstructures on wear resistance were studied; and a large base of data is given.

Larsen-Basse (1968a & 1968b) also carried out work to study the effects of grit size on abrasive wear by varying emery papers and loads. The results show rapidly increasing wear with grit size until a 'critical size' was reached. By studying the material removal with respect to the applied load, the author deduced that some grains must be supporting load without cutting. He suggested that for fine grits as they breakdown more load is supported elastically; for coarse grits the breakdown has less effect on the cutting area (the larger grains are further from the backing paper and fracturing away the tops still leaves cutting surfaces). The author measured the number and size of grooves formed; he found a large range of widths and estimated that some 50% of the grits were cutting. He also estimated that 15% of the groove volume was removed as a chip, the rest being formed into ridges on the surface.

Buttery and Archard (1971) studied the grooves formed during scratch tests and from abrasive grinding. The geometry of the scratches were measured and classified in terms of three areas A_1 , A_2 , and A_3 as shown in Figure 6.4.

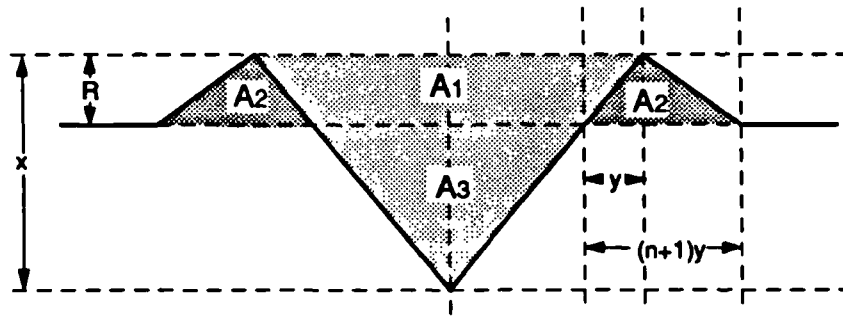


Figure 6.4. The classification of abrasive wear scratches by Buttery and Archard (1971).

The scratch volume is given as $(A_3 + A_1)$ and the proportion of material removed as wear, β was suggested to be $(A_3 - A_2)/(A_3 + A_1)$. The wear equation of the type developed by Rabinowitz (equation 6.4) was then proposed;

$$\frac{V}{l} = \frac{0.5\alpha\beta W \cot \theta}{H} \quad (6.6)$$

where α is the proportion of grits cutting, the other variables having the same definition as those in equation (6.4). The authors performed some repeated scratch tests and found that the proportion of material removed increased with the hardness of the abraded surface, with softer materials the bulk of the groove material was piled-up as shoulders.; their data is shown in Figure 6.5.

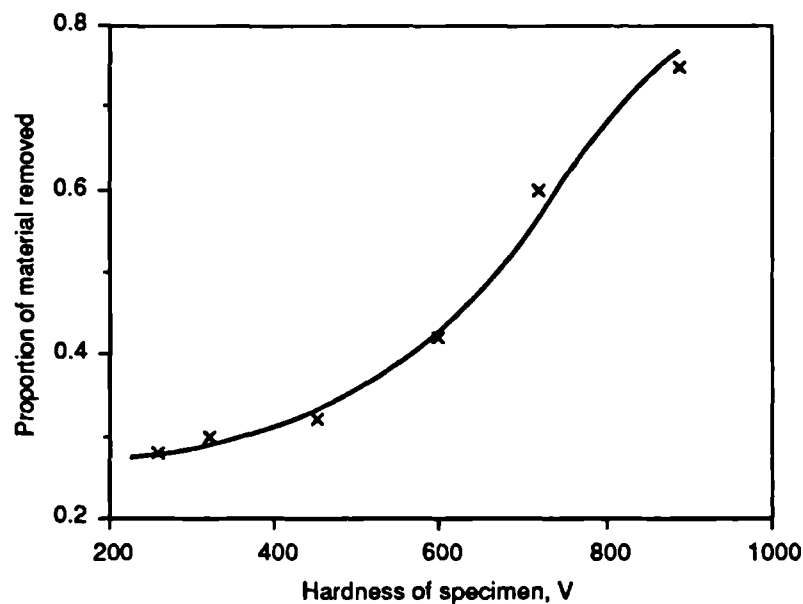


Figure 6.5. Proportion of material removed from a groove as a function of specimen hardness. Data extracted from Buttery and Archard (1971).

Challen and Oxley (1979) developed slip line field models (originally from the work of Lee and Shaffer (1951) on machining processes) to determine the regimes of plastic flow around a wedge shaped sliding asperity. They determined the regimes of wave flow and chip removal which depended on the asperity slopes, interfacial shear strength, and yield stress in shear.

The elastic effects in abrasive wear were investigated by Torrance (1980a). He noted that the typical abrasion model, based on the geometric measurements of a scratch (such as that of equation 6.6), took no account of the elastic distortion of the surrounding material. The elastic deformation was calculated and included in the geometry of the scratch. The wear volume was then determined from the number of ploughing grains, sliding distance, fraction of the groove removed as wear, and the corrected groove shape. Torrance used Richardson's data to determine the constants in his equation, and then demonstrated good between his theory and the data of other investigators.

Torrance (1980b) also performed scratch tests and noted a phenomena of abrasive material removal from the side walls of the groove. He used a slip line field method, similar to that of Challen and Oxley, to model the process. Predicted forces and wear volumes were found to agree well with the experimental data. In a later short communication (Torrance 1981), he used a slip line field indentation model of an asperity/metal contact to predict asperity collapse. He showed that when the hardness of the abrasive asperity was approximately 1.25 that of the material, then wear becomes significant. This finding was in agreement with the experimental evidence of Richardson.

Murray et al (1982) used the results of pin on disk scratch tests to determine critical attack angles for cutting. The authors point out that only a certain number of the grits on abrasive papers have cutting angles greater than the critical value (see Figure 6.6); so only a certain proportion will actually cut.

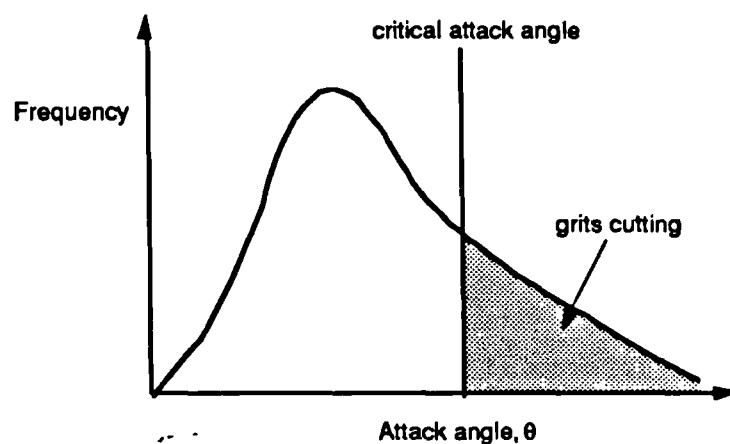


Figure 6.6. Schematic representation of the distribution of grit attack angles in unused silicon carbide paper. Only those with an attack angle greater than some critical value produce a wear chip.

The critical attack angles were estimated as 55°, 85°, 45°, and 65° for lead, aluminium, copper, and nickel respectively. The publication includes some interesting SEM photographs of the resulting scratches which show the mechanisms of ploughing, cutting, and material fragmentation.

Generally, the indentation of harder materials causes more pile up around the indenter (see §4.4.3). This can then be more easily removed by the relative sliding of the indenter. The fraction of material removed is therefore greater and the critical cutting angle lower. The process is therefore attributable to the work-hardening behaviour of the material (i.e. hard materials are frequently already work hardened). Figure 6.7 shows the reduction in f factor for materials of increasing work hardening rate. The data is extracted from Hokkirigawa, Kato, and Li (1988).

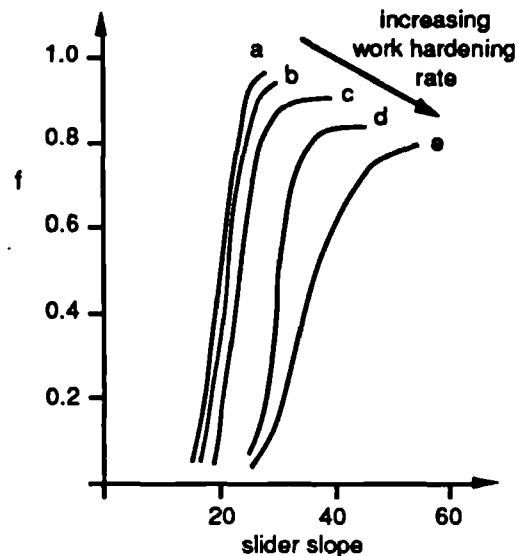


Figure 6.7. The fraction of the scratch volume removed as a wear chip, f , against the slope of the slider. The five curves (a to e) are for increasing work hardening rate steels.

Kato et al (1986), performed a comprehensive series of scratch tests in an SEM. The studies thoroughly cover the transitions between ploughing, wedge, and cutting flows for spherical tipped and three dimensional pyramidal sliders. The publications systematically maps out the wear modes on plots of slider geometry and attack angle. Figure 6.8 shows such a map of the regimes of flow around a pyramidal slider.

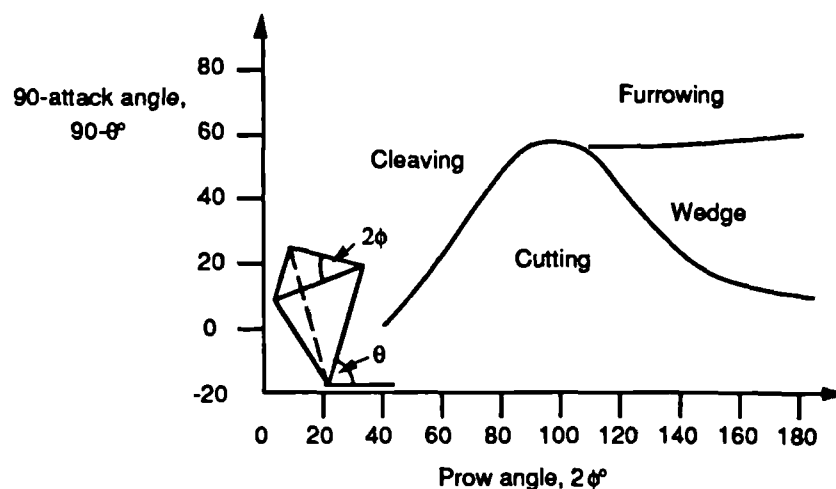


Figure 6.8. Flow regimes around an edge forward pyramid cutting soft aluminium. 90° minus the attack angle, θ is plotted against the prow angle, ϕ of the pyramid. Regimes of wave, wedge, furrow, or cutting chip formation occur. The slider angles are defined in the inserted sketch.

A study of the role of lubrication in abrasive wear was performed by Torrance and d'Art (1986). Lubricated wear rates in pin on abrasive paper experiments were as much as three times higher than in dry conditions. The results were in keeping with the predictions of Challen and Oxley that low friction encourages cutting flows. Also in dry conditions abrasive clogging was found to occur rapidly.

Zum Gahr (1987) gives a theoretical model for the prediction of the proportion of material removed from a groove, f_{ab} . The form of the relation is;

$$f_{ab} = 1 - \left(\frac{\phi_{lim}}{\phi_s} \right)^{\frac{2}{\beta}} \quad (6.7)$$

where ϕ_{lim} is a parameter describing the effective deformation of a surface, and ϕ_s is the capability of deformation of the material, and β is a factor describing the decay of deformation with depth, mainly due to the work hardening behaviour of the material. He used fracture mechanics to predict this removal factor for brittle materials where chips may flake from the sides of a groove. His wear model includes these parameters and factors for the abrasive grit shape. Zum Gahr (1988) includes a comparison of model predictions and experimental results. The material removal factor agrees well with predictions for a wide range of alloys and pure metals. The wear resistance of these materials was found to correlate well with the ratio of hardness (after being fully deformed) to f_{ab} .

This review is presented to assist in the understanding of the three-body rolling bearing type of wear mechanism. As such, it is not intended to be a thorough investigation into two-body abrasive wear. If such a review is required, the reader is directed to Childs (1988) who has compiled a detailed comparative survey in this field.

6.6. Three-Body Abrasive Wear

Rabinowitz, Dunn, and Russell (1961) carried out the first experiments into three-body abrasive wear, using apparatus consisting of a flat specimen rotating on an annular specimen fed with a column of abrasive grit. The authors found the abrasive wear resistance to be approximately proportional to specimen hardness. Also, as noted by Kruschov, they found a less than expected wear resistance for heat treated steels. The authors experimented to determine the effect of abrasive grit size on wear rate. It was found that for small grit sizes wear rates were low, as the grit size increased above about $45\mu\text{m}$, wear became independent of grit size. A table of abrasive wear constants (i.e. the slope of the wear resistance/hardness graph) was given comparing the results of data for two and three-body wear for various abrading materials. It was found that three-body wear coefficients were an order of magnitude smaller than two-body. This was explained by the free grits rolling for much of the time and only causing wear when sliding occurs.

Rabinowitz and Mutis (1965) investigated further their earlier findings of a threshold abrasive size above which wear does not increase for increasing particle size. The authors suggest that when fine grits are used some two-body contact occurs and adhesive wear particles are formed. These were then thought to interfere with the three body abrasive wear process. The authors carried out abrasive experiments using silicon carbide powder on copper specimens. Critical size cut offs were again found to be about $45\mu\text{m}$ for copper and about $55\mu\text{m}$ for nylon. As a comparison both oil and water lubrication was employed; the critical size effect did not occur in the water case. It was suggested that this was because significant adhesive wear masked the effect. The authors used surface energy analysis to predict the size of these interfering adhesive particles; the size compared reasonably well with the critical grit sizes.

Broeder and Heijnekamp (1966) used a journal bearing for their experiments on abrasive wear. A hard steel shaft was used with a variety of bush materials. Wear rates were found to be constant; since, the geometry of the contact was changed little during the test. Soft bush

materials did not generally lead to lower shaft wear; abrasive particles were found imbedded in the bush surface which repeatedly scratched the shaft. The study included a close inspection of the worn surfaces. Pitting type wear was observed, where large flakes became detached from the shaft. Ploughed furrows, ridges, and cutting wear were observed.

The effects of anti-wear additives on three-body abrasion was studied by Tao and Appeldorn (1969). A vane pump was used with lubricants dosed with oleic acid or stearyl amine and $1\mu\text{m}$ iron powder or coarse silica abrasives. Wear was measured as the mass loss of the pump vanes. At low pump loads it was found that the additives eliminated the abrasive wear completely; at higher loads the additives were only partially effective. It was explained that the chemical action between correctly matched additive and abrasive (i.e. oleic acid with basic iron oxide or basic amine with acidic silica) was most effective at reducing wear rates. It was also found that pre-coating the abrasives in the additive also reduced wear, proving that the wear reductions were not caused by protective film formation on the pump vanes. It was also suggested that wear reduction may be caused by the additive reducing the ability of the particles to adhere to one surface and therefore abrade the other.

Larsen-Basse (1974) extended the analysis to include the effects of humidity on three-body abrasive wear; using silicon carbide and alumina particles to abrade metal cylinders pressed against a backing wheel. Humidity was found to rapidly increase wear rate. This was explained by, amongst other things, the moisture aiding the process of grit fracture which left sharper cutting surfaces.

Miller (1980) reported some closed three-body abrasive wear results for various sizes of diamond particles abrading molybdenum alloy spheres sliding in cups. The authors noted increasing wear rates with increasing abrasive particle size up to a critical size. This critical size was in agreement with the measurements of Rabinowitz (approximately $50\mu\text{m}$). Wear was found not to occur with $1\mu\text{m}$ particles since the size was below the contact film thickness. The authors noted no large size adhesive wear fragments in the resulting slurry and so disputed the Rabinowitz explanation of the critical size effect. It was also noted that the critical size effect became less pronounced at higher loads.

Misra and Finnie in a series of papers (1980, 1981a, 1981b, 1983) performed many three-body tests with columns of abrasive pressed onto a rotating metal disk. They compared the processes of two and three-body abrasive wear. Both were found to show the critical size effect and the lower than expected wear resistance found in work hardened materials. They compared the hardness differential required for abrasive wear and found similarities. However, it was noted that three-body wear varies non-linearly with load at low loads and linearly at higher loads; whereas the two-body variation is linear. This was explained by the lower degree of constraint experienced by particles in three-body abrasion at low loads.

They reviewed all previously proposed theories on the two and three-body abrasive grit size effect. They discounted several; abrasive clogging, abrasive grit damage, elastic contact, adhesive wear particles, particle attack angle, and strain rate effects. The authors believed that the only rational explanation of the size effect was the higher flow stress found at the surface of the abraded materials. Small particles act in this layer, whilst the larger particles will abrade through it. Hence, increasing the particle size will have no greater abrasive effect.

Their work also included a scheme of three-body abrasion experiments which comprehensively investigated many of the variables in the process. Wear was found to increase linearly with path length after an "incubation period" where mass increased. This mass increase was caused by the imbedment of the abrasive grains in the test surface. The incubation period was longer for softer materials. Also, wear was found to increase linearly with load. But wear rate was found to vary little with sliding speed. The next variables tested were the hardness of the abrasive and

specimen; wear resistance was found to vary linearly with hardness (as noted by previous workers), and only abrasives with a hardness 1.25 times greater than that of the specimen were found to cause significant wear. In their experiments, the mechanism appeared to be one of indentation, when sharp edge grits roll on the surface, and little ploughing of grooves. The wear mechanism was largely ascribed to the indentation process and the formation of lips which later are separated from the surface by the action of further particles.

Sarkar (1981) presented the findings from SEM studies of three-body silicon carbide abraded brass. He noted grooves cut in the softer surface and the formation of hills and valleys parallel to the direction of motion. The hills were frequently abraded by rubbing with the counterface. The valleys were found to wear by the plucking of imbedded abrasive grains. A process of micro-spalling was also found to occur after repeated abrasive action.

Test on abrasive wear in hydrodynamic bearings were carried out by Ronen et al (1980, 1981, 1983). A contaminant mix consisting of Arizona dust, alumina, silicates and iron oxide was used with babbitt or copper alloy bearing liner. The authors noted rapid increase in wear when the abrasive grit sizes were greater than the film thickness. The relative distribution of wear between the shaft and liner were found to be dependent on the hardness ratio. Shaft wear increased as its relative hardness increased; at high hardness ratios (from 1:5 to 1:15) the wear rate stabilised. The abrasive particles were believed to imbed in the softer surface and plough the harder.

An interesting aspect of the previous study was highlighted during examination of the surfaces. A mixture of ploughed grooves, and indentation marks were present. The wear was thought to be a combination of grit ploughing and tumbling action. A model was presented which predicted the critical particle size to film thickness ratio, $(D/h)_{cr}$, required for particles to lock in the contact and cause ploughing;

$$\left(\frac{D}{h}\right)_{cr} = \frac{1 + \tan^2 \theta \left\{ 1 + \sqrt{\left(\frac{H_1}{H_2}\right)} \right\}}{1 - 0.5 \tan^2 \theta \left\{ 1 + \sqrt{\left(\frac{H_1}{H_2}\right)} \right\}} \quad (6.8)$$

where H_1 and H_2 are the hardnesses of the two materials, and θ is an abrasive grit angle. For ratios less than about 2.5 to 4 (depending on particle geometry and hardness ratios) the grits were expected to lock in the contact and score the harder body. The test conditions where particle locking was observed correlated well with this theory.

Odi-Owei and Roylance (1986) noted the lack of data for three-body wear in concentrated contact situations and performed some experimentation on abrasive wear in a four ball test machine. Their test were performed under different sliding lubrication regimes. Under conditions of ehl, abrasive ploughing occurred with particles imbedded around the contact inlet, sometimes blocking oil flow; wear and friction were found to increase with both concentration and particle size. Under conditions of boundary lubrication the abrasive effect was generally masked by adhesive and scuffing processes. Although, it was noted that under boundary lubrication, abrasion left a rough surface from which gross scuffing initiated. A simple friction force model (similar to that derived in §2.4.4.1) is presented to explain the relative size of particle which may be entrained into the contact for various coefficients of friction.

Mehan (1988) aimed his work towards finding material combinations to resist abrasive wear in applications such as a piston ring and cylinder liner. Aluminium oxide was used as an abrasive

in pin-on-disk wear experiments. A wear rate ranking of many materials was produced; the results were generally dependent on hardness properties. Small size abrasives $0.3\mu\text{m}$ were found to cause negligible abrasive wear even though the film thickness was estimated as $0.05\mu\text{m}$; the authors suggested that these small particles had a more rounded, less damaging, morphology. Ceramic materials, tungsten carbide, silicon carbide, silicon nitride and alumina were found to be the most wear resistant.

Xuan, Hong, and Fitch (1989) were also interested in the wear of tribological pairs under abrasive conditions. They used journal bearings of different materials with various grits to obtain a range of surface hardness ratios (H_b/H_j) and abrasive to surface hardness ratios (H_j/H_a). Results showed that harder abrasives caused greater wear; the wear rates dropping off with time because particles broke down and blunted. Journal wear was found to decrease with increasing journal-to-abrasive hardness ratios and to increase with increasing bearing-to-journal ratios. The critical hardness ratio, for a particle to imbed in one surface and not damage the other, was investigated. It was found that the lower the value of H_b/H_j the lower the wear and the critical H_j/H_a .

Williams and Hyncica (1992) performed abrasive wear experiments using abrasives in a foil bearing. In their experiments it was noted that when the ratio of grit size to film thicknesses was low the particles appeared to be tumbling leaving small erosion pits; whilst at higher ratios grooving occurred. They suggested that the critical particle size to film thickness ratio for this to occur was given by;

$$\left(\frac{D}{h}\right)_c = \sec \beta \quad (6.9)$$

where β is an abrasive grit angle. This transition typically occurred for ratios of about two. Using the force equilibrium and geometry of a single abrasive grit they modelled the tumbling and ploughing processes and show the conditions for a particle to be trapped in the soft body whilst cutting the harder body. The model also demonstrated how a harder material may wear more than its softer counterface.

Kragelsky, Dobychin, and Kombalov (1991) recognise the importance of defining the motion of the particle between the two surfaces. They use the parameters α and β to express the velocity of the particle, v as a ratio of the velocities of the two surface, v_1 and v_2 . So;

$$v = \alpha v_1 + \beta v_2 \quad \text{where} \quad \alpha + \beta = 1 \quad (6.10)$$

They determined the ratios α and β to be inversely proportional to the hardness of the two surfaces. (This is shown to not be generally applicable; in the tests described in the next chapter particles were found to imbed in one surface and scratch the other). Hence;

$$\alpha = \frac{H_2}{H_1 + H_2} \quad \text{and} \quad \beta = \frac{H_1}{H_1 + H_2} \quad (6.11)$$

The model is then developed by considering spherical abrasive particle travelling at this velocity relative to each surface. Deformed volumes are calculated from the geometry of the particle, which is considered to fragment in the contact. The wear is calculated as a multiple of the deformed volume for each particle fragment.

Fang, Kong, and Zhou (1992) developed an apparatus to study the motion of abrasive particles trapped between two surfaces undergoing relative motion. 200-300µm particles of sand were interposed between cast iron surfaces, and their motion studied when the upper surface was moved across the lower. The sharper sand particles were found to be more likely to tumble whilst the rounded particles more likely to plough. The former mechanism causing scratches whilst the latter left pits on the iron surfaces.

6.7. Wear in a Rolling Element Bearing

A few studies have been specifically aimed at abrasive wear in rolling element bearings. A brief review of these publications is presented here. The test results are discussed further in §7.4, where they are compared with the results of the bearing tests in this study.

Fitzsimmons and Cleveger (1975) studied the wear of roller bearings with contaminated lubricant. The source and flow path of debris is discussed and they recognise that particle size reduction must occur. Tests on roller bearings were performed using silica as an abrasive; wear was found to be proportional to the abrasive concentration. Increasing the particle size generally caused an increase in wear. Abrasives of a range of hardness were tested; the harder material caused more wear. However steel particles were not as damaging as silica dust of similar hardness, presumably because they deformed in the rolling contacts. Mass losses were found to level off after test durations of about 10 hours, as the particles become reduced in size. The addition of further contamination caused a new cycle of wear.

The study also included some experiments varying the lubricant base for the tests. Generally more viscous oils, generating a thick film, caused less bearing wear. However, an increase in wear was noted when EP additives were present in the lubricant. It was suggested that the abrasive action generated a fresh metal surface each time which was then exposed to chemical attack from the additive; the two processes in tandem lead to increase material removal.

Loewenthal, Moyer and Needelman (1979 & 1982) performed a series of experiments on the effectiveness of filtration on ball bearing lives. Lubricants were contaminated with a mix of graphite 1%, Arizona dust 11%, and steel particles 88%; the debris mix was based on findings from the analysis of lubricants from aircraft gas turbines. The tests were aimed at investigating fatigue failure; but in one series, using coarse filtration, excessive wear was noted. The test bearings had lost an average of 1g of material in 448 hours. Photomicrographs of the bearing surfaces show a heavily burnished surface.

Perrotto et al (1979) ran ball bearings with lubricant contaminated with silica or alumina particles. They measured the electrical resistance as a measure of the asperity contact. Increasing both, the size of the abrasive particles, and the concentration, was found to cause more contact. It was also reported that when smaller particles were used, the bearing was able to run-in to a lower level of contact. Both steel and nylon cages were tested; the worn surfaces of the balls were different under these conditions. The nylon caged balls appeared to have pits on the ball surface whilst the steel caged balls were scratched and scored.

Skoryin and Minchenya (1984) developed a model for bearing wear based on the motion of the cage. The authors note that the textolite cage material becomes charged with debris, and attribute bearing wear to this action. The abrasive wear between the rolling elements and raceways was thought to be negligible²⁰. The model was based on cage loads and sliding

²⁰ The experiments detailed in this thesis show this assumption to be generally not valid. No evidence of cage charging was noted, and weight loss distribution on the bearing surfaces were found to be dependent on the microslip in the ball/raceway contact.

distance probabilities, including a statistical term for debris entering a contact. The wear equation was of a similar form to the Archard expression; with mass loss proportional to sliding distance, load, and number of entrained particles. To verify the model, bearings were contaminated with corundum particles; doubling the particle size caused an increase in wear by a factor of four. In the tests the balls wear more than the races (72% ball wear, 10% inner race, 3% outer race, and 15% cage). No direct comparisons are made between the theory and the test results.

The FAG catalogue²¹ (1981) gives a scheme for predicting bearing wear life for their products. The user first determines a factor, from 'a' to 'k', for the operating conditions from an application table. For example, the least favourable operating condition, designated *k*, occurs in motor vehicle gear boxes, axle drives, and belt conveyors. The user then calculates a wear factor, *f_w*, from the bearing bore size and the amount of clearance tolerable in the application. Given these factors the user is then directed to a chart of wear factors against bearing life for the range of operating conditions *a* to *k*. For the worst operating conditions lives range from 300 to 3500 hours. The calculation is simple to perform and provides wear data (or application experience) for the benefit of the user. Another useful feature of the method is that information can be obtained on the tolerable clearance in an application. For example in a motor vehicle gear box a wear factor in the range 5-8 is quoted, this implies a permissible clearance in the range 20-32μm.

6.8. Closure

A survey of the literature on wear in rolling bearings and other closed three body applications is presented. Surprisingly little material seems to be available, particularly concerning the fundamental mechanisms of the processes involved.

Much data exists on the two body wear process, from cutting tool and abrasive paper studies. Useful information about relevant parameters can be obtained from this source. Particle size, relative hardnesses, lubrication, cutting angles, load, sliding distance, and speed are important factors.

²¹ The method appears in the 1981 edition but has been omitted from later copies.

CHAPTER 7

ABRASIVE WEAR EXPERIMENTS

7.1. Introduction

The survey of published work on three body abrasive wear has shown some of the parameters which may be important in controlling abrasive wear; particle size, relative hardness, sliding speeds, and load. It is perhaps surprising that little work has been done to understand the fundamental mechanisms of abrasive wear by particles trapped in a concentrated contact. Also, little published material on the three body abrasive wear of rolling bearings has been found.

This chapter describes two sets of experiments. The first set uses a simple ball on disk geometry, in an attempt to understand some fundamental aspects of three body abrasive wear. Particle size, debris material, rolling element hardness, slide/roll ratio, and lubricating film thickness were varied. In each case the nature of surface damage was used to investigate the mechanisms of abrasive wear would develop.

The second set of tests use a bearing test rig. The experiments were performed to quantify the abrasion process in ball bearings and relate these back to the mechanisms of particle entry, debris damage, and wear.

7.2. The Mechanism of Abrasive Wear

One of the fundamental requirements for understanding the mechanisms of three body abrasive wear is a knowledge of how the particle behaves in the contact. The optical ehl rig was useful in the study of particle motion and breakdown, but could not provide information on behaviour between two sliding steel surfaces.

To provide this information a simple test geometry of a ball on disk was used. This allowed single parameters to be tested in isolation; speed, load, film thickness, materials, lubricants, debris size and material could all be varied easily. Examination of the damaged surfaces provided a tool for quantifying particle motion in the contact. This was then be related to the abrasive wear mechanism.

7.2.1. The Ball on Disk Machine

This test configuration consisted of a 1" ball, loaded onto a cylindrical disk. The ball was supported by three small ball bearings. A crossed groove was cut into each ball for location on the drive shaft. The disk was held on its shaft by a taper lock. Both shafts were mounted using deep groove ball bearings into housings. The ball shaft housing was fixed rigidly to the frame of the rig; whilst the disk shaft housing was free to pivot. A cantilever was fixed to the pivoting housing; dead weights were applied to the cantilever to achieve the required contact load²². Figure 7.1 shows an elevation of the rig with the fixed and pivoting shaft housings.

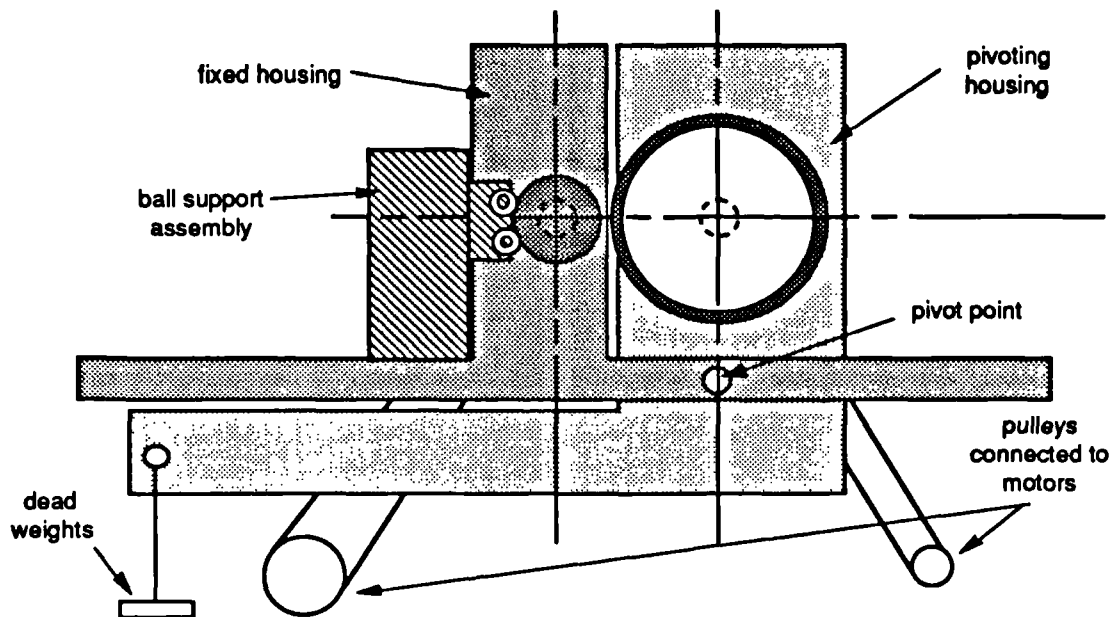


Figure 7.1. Schematic elevation view of the ball on disk test machine; showing specimens, shaft housings, and ball support.

The ball was mounted on a hollow tubular insert which screwed into the drive shaft (see Figure 7.2). Four strain gauges were mounted on the insert in a rosette; the four gauges were wired in a Wheatstone bridge arrangement with their axis at forty five degrees to the insert axis. The wires passed through a hole in the tube and were connected to slip rings mounted on the end of the shaft. These terminals were then connected to a transducer indicator (RHP E307-2) and the output signal from this passed to the analogue to digital port of a BBC microcomputer. This arrangement gives a method for measuring the torque transmitted to the ball.

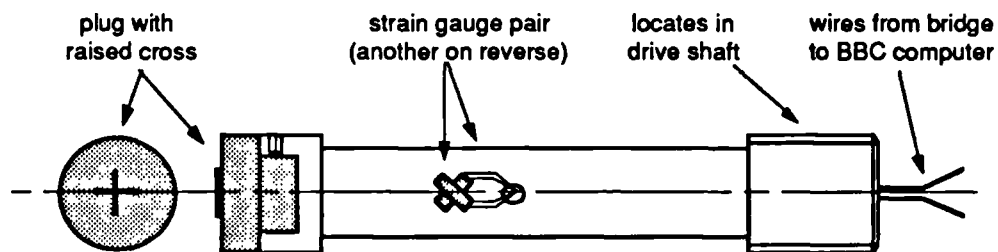


Figure 7.2. Schematic view of the ball shaft insert or 'torque tube'.

²² The rig has a design fault associated with this loading system. Friction forces transmitted across the contact cause a moment about the housing pivot and will lead to an increased contact load. However, friction coefficients are low and so this added load is small compared with the applied load.

Optical counters were used to determine the rotational speed of the two shafts. The connections of the optical counters (a 5V supply, earth, and signal) were connected to the user port of the BBC microcomputer.

A simple computer program was written to record the signal from the torque meter and the number of pulses from the optical counters, and display the rotational speeds and torque on the microcomputer screen. The output from the torque meter was calibrated by applying a range of known couples using a weighted beam. The measured torque was reasonably accurate when first calibrated (within 10-15%). The accuracy tended to drift with time, possibly as a result of the build up of spurious strains in the shaft, frequent re-calibration was required.

Both shafts were driven through an arrangement of pulleys and belts by two separate electric motors. These motors were controlled independently to achieve any required slide roll ratio at the contact. Figure 7.3 shows a top view of the shaft assembly with the pulleys, specimens, and ball support unit. A sealed oil bath surrounded the two specimens; lubricant was fed directly into the constriction and mixed by means of a vigorous electric stirrer and peristaltic pump recirculating the flow.

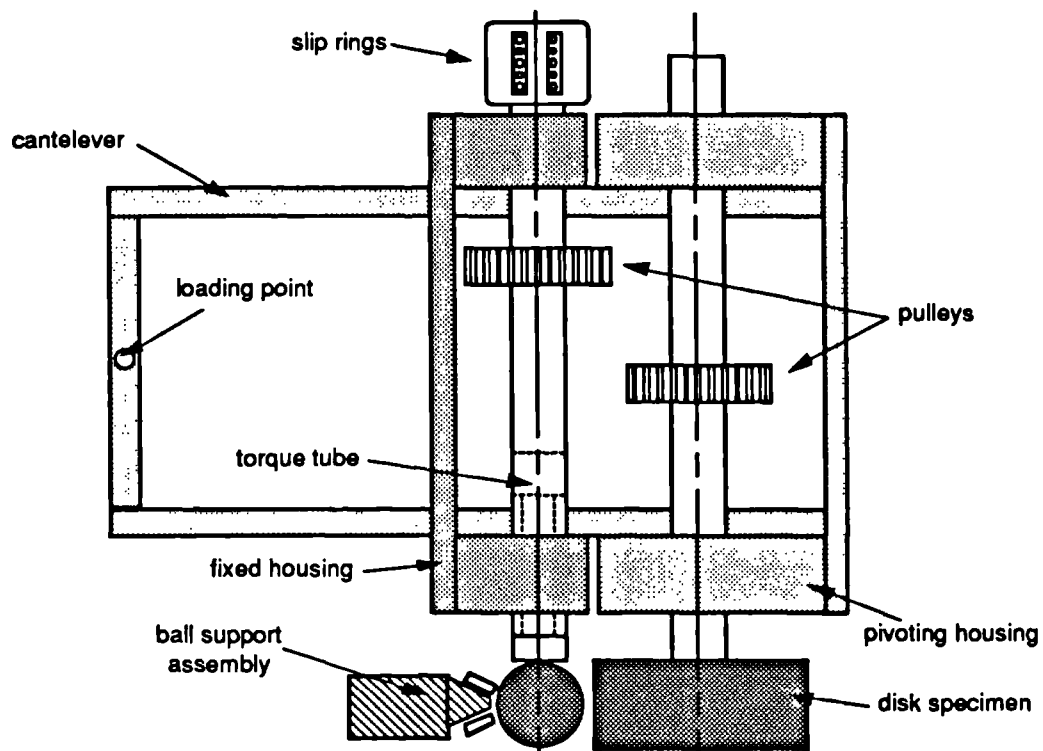


Figure 7.3. Schematic top view of the ball on disk test machine; showing shafts, housings, pulleys, and ball support.

The photographs included as Figures 7.4 and 7.5 show the side and top views of the apparatus.

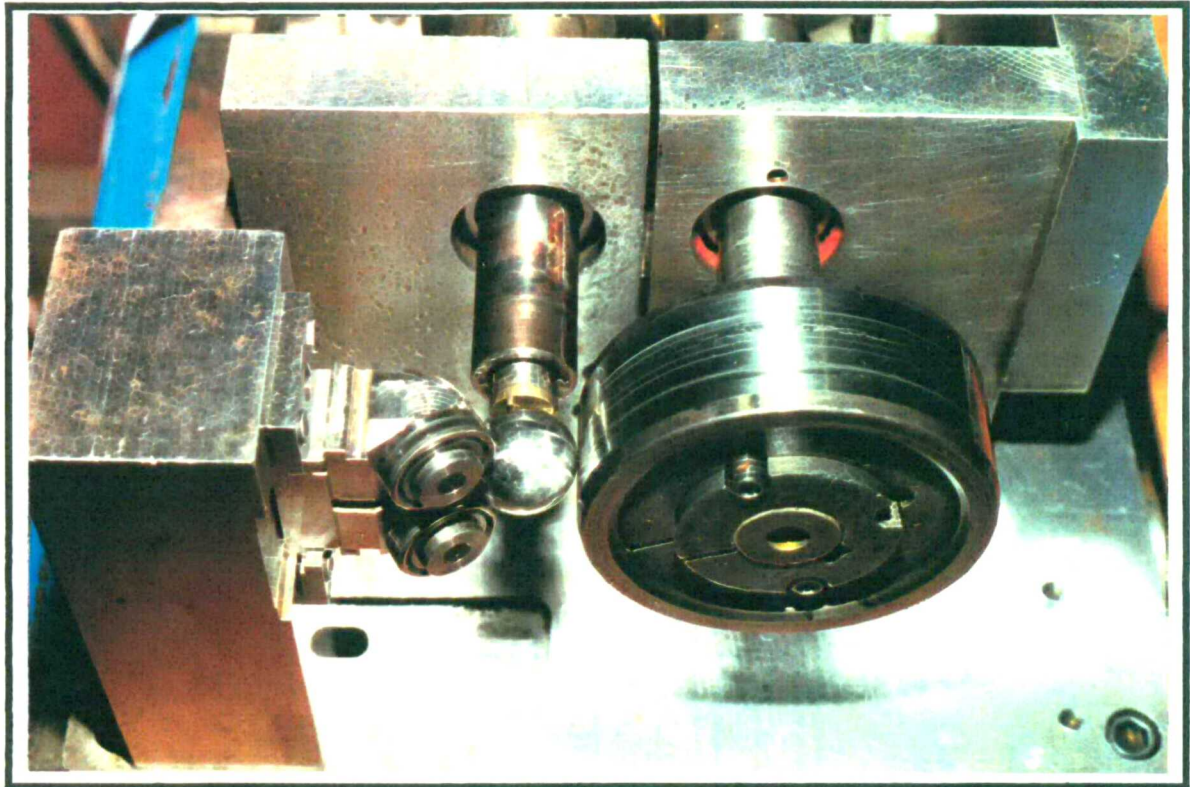


Figure 7.4. Photograph of the side view of the ball on disk test machine; showing specimens, ball support, and housing ends.

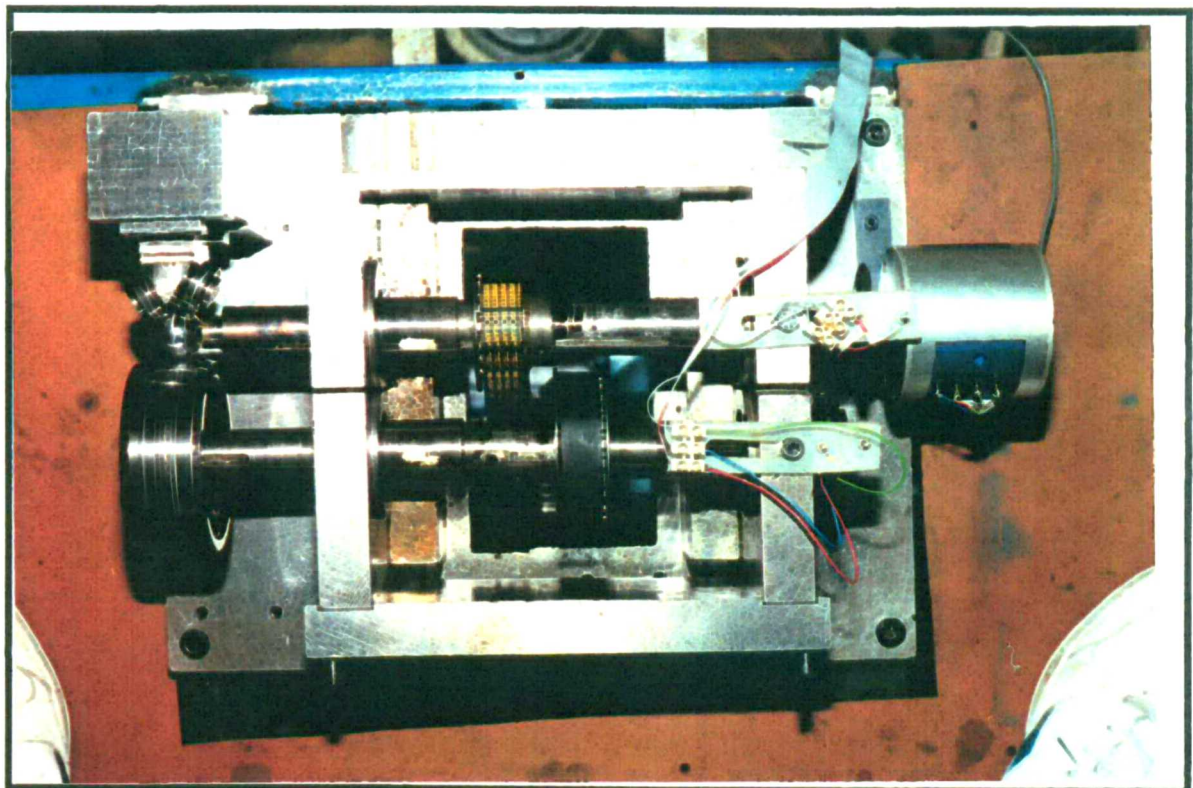


Figure 7.5. Photograph of the top view of the ball on disk test machine; showing shafts, housings, pulleys, slip rings, and ball support.

7.2.1.1. Test Specimens

The test specimens were components from standard rolling bearings. The balls were 1" diameter from a deep groove ball bearing. The disks were inner raceways from a cylindrical roller bearing (full details of the specimens can be found in Appendix 1). Slots were cut in the balls for driving purposes.

For some of the tests it was necessary to soften the specimens. This was done by coating them in a oxidation resistant paint, heating them in a furnace, and air cooling (see Appendix 1 for details).

For some of the tests silicon nitride balls were used; these specimens had been manufactured specifically for research purposes for SKF ERC. They provided a very hard specimen as an extreme data point.

7.2.1.2. The Test Programme

The test programme consisted of six series of experiments designed to investigate some of the fundamental mechanisms of three body abrasive wear. In many of the tests 6 μ m silicon carbide particles were used; from previous experience (see section §2.4) it was noted that these particles were unlikely to fracture in the contact. The abrasive wear mechanism would then be controlled by a particle of (almost) known size and shape.

The test programme was not firmly devised in advance, rather, as the results of one series were analysed it became desirous to investigate another aspect. The final test series performed were as follows;

- (a) The effect of slide/roll ratio. The same debris particles (6 μ m silicon carbide were chosen since these would not fracture) were mixed with the lubricant and fed into contacts of increasing slide/roll ratio, from 0% to 40%. All other parameters remaining the same.
- (b) The direction of sliding. The above tests were repeated but with the direction of sliding reversed.
- (c) The effect of particle size. The debris material used was synthetic diamond in five different size bands (0-0.5 μ m, 0.5-1 μ m, 1-2 μ m, 2-4 μ m, and 3-6 μ m). All other parameters remaining the same.
- (d) The effect of debris concentration. The concentration of 6 μ m silicon carbide particles was increased (from 0 to 2000mg/l) for a pure rolling test. All other parameters remaining the same.
- (e) The effect of rolling element hardness. The hardness of the specimens was varied by tempering; silicon nitride balls were also used.
- (f) The effect of debris material. A range of other debris materials (low carbon steel, Arizona dust, alumina, and copper) were also tested.

7.2.1.3. Experimental Method

The abrasive material to be tested was mixed thoroughly with the test lubricant (Shell Turbo T68, see Appendix 1 for details). A concentration of 25mg/l was used, higher concentrations

tended to cause excessive surface damage and obscure the nature of the abraded grooves. An even mixture was obtained by forming a paste of powder in oil and gradually diluting. The mixture was then poured into the lubricant bath and the stirrer motor and circulating pump started.

The motors were switched and the required slide/roll ratio set²³. The contact load was then applied (since the tests were for a short duration it was necessary to apply load only when the specimens were up to speed; otherwise a good proportion of the test may have taken place at a reduced speed). The load was abruptly removed at the end of the tests. Ten minutes of running time (about 20,000 ball revolutions) was found to be sufficient to generate an easily visible number of surface grooves.

As the disk was 40mm wide; it was possible to fit several contact tracks on each disk, whilst the ball was replaced after each test. Appendix 1 gives details of the contact dimensions.

7.2.1.4. Test Measurements

The tests were aimed at understanding the mechanisms of particle controlled wear. The test measurements reflect this desire to visualise the wear process.

- (i). Visual examination of the surface. The surfaces of both test specimens were examined under an optical microscope. The size and shape of surface scratches were noted. The width of the contact track was recorded. Photographs of relevant subjects were taken.
- (ii). Surface profile of the worn region. A Form Talysurf was used to record surface profiles of the worn region. The shape of surface scratches was investigated by magnifying individual grooves or scratches. Generally the shapes of the scratches were of shallow enough slope so that the Talysurf stylus did not flank-out and was able to penetrate to the groove bottom. The scratches were sometimes only just discernible from the surface roughness of the specimens²⁴. In these instances the profiles were filtered to remove the roughness element; a low pass filter of 2 μ m was found to remove the roughness but leave the abrasive groove largely intact. The process of scratch measurement was somewhat statistical; many profiles were taken in order to achieve an appreciation of the groove size.
- (iii). Torque Transmitted to Ball. The reading from the torque meter was recorded several times during each test. Traction coefficients were calculated from the torque and the applied load.

The test arrangement made the measurement of mass loss of little value. The ball specimen was supported by three ball bearings all of which have their own abrading contact track and will contribute to a mass loss. The test durations were short (of the order of ten minutes) and so mass loss would be low. The disk specimen was too heavy to weigh with the required accuracy (i.e. to ± 1 mg).

²³ In some slide/roll combinations the torque transmitted to the ball tended to overpower that from the motor and the contact would jump into pure rolling. It was necessary therefore to introduce a tensioning wheel onto the disk shaft to reduce its energy.

²⁴ This was particularly the case for the dents on the disk surface. The disk had a rougher surface finish and the results were generally difficult to interpret.

7.2.2. Results

Some of the tests contributed to an understanding of the role of more than one parameter. For clarity, therefore, it is easier to discuss the results of these tests under the heading of the relevant parameters. The parameters; slide/roll ratio, traction, particle size to film thickness ratio, concentration, and materials hardness, will be discussed in turn.

7.2.2.1. The Shape of the Abrasive Groove

By extracting and expanding sections of surface profiles across the wear track of the balls, it is possible to investigate the size and shape of single abrasive grooves. Some typical profiles of sections through grooves are shown in Figure 7.6.

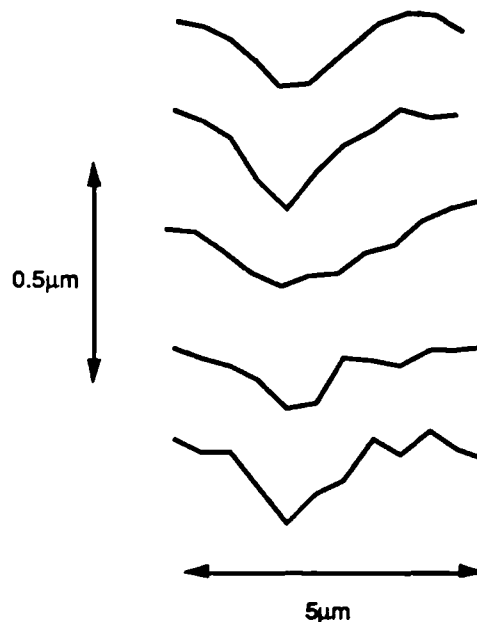


Figure 7.6. Surface profiles of abrasive grooves (by $6\mu\text{m}$ silicon carbide particles) recorded as sections through the groove perpendicular to the direction of sliding.

Profiles of the surface scratches show the presence of slightly raised shoulders at the edges. Statistical measurement of many of these grooves has shown a similar form for most grooves. The scratch definition parameters used by Buttery and Archard (1971) and displayed in Figure 6.4 were used here. Typically the slope of the grooves is about 10° , and the shoulder heights about 10-20% of the groove depth. The scatter in this data is discussed in the §7.2.2.6.

The photographs included as Figure 7.7 show typical worn ball surfaces for a range of slide roll ratios. The length of the groove is close to the debris sliding distance (see §7.2.2.2).

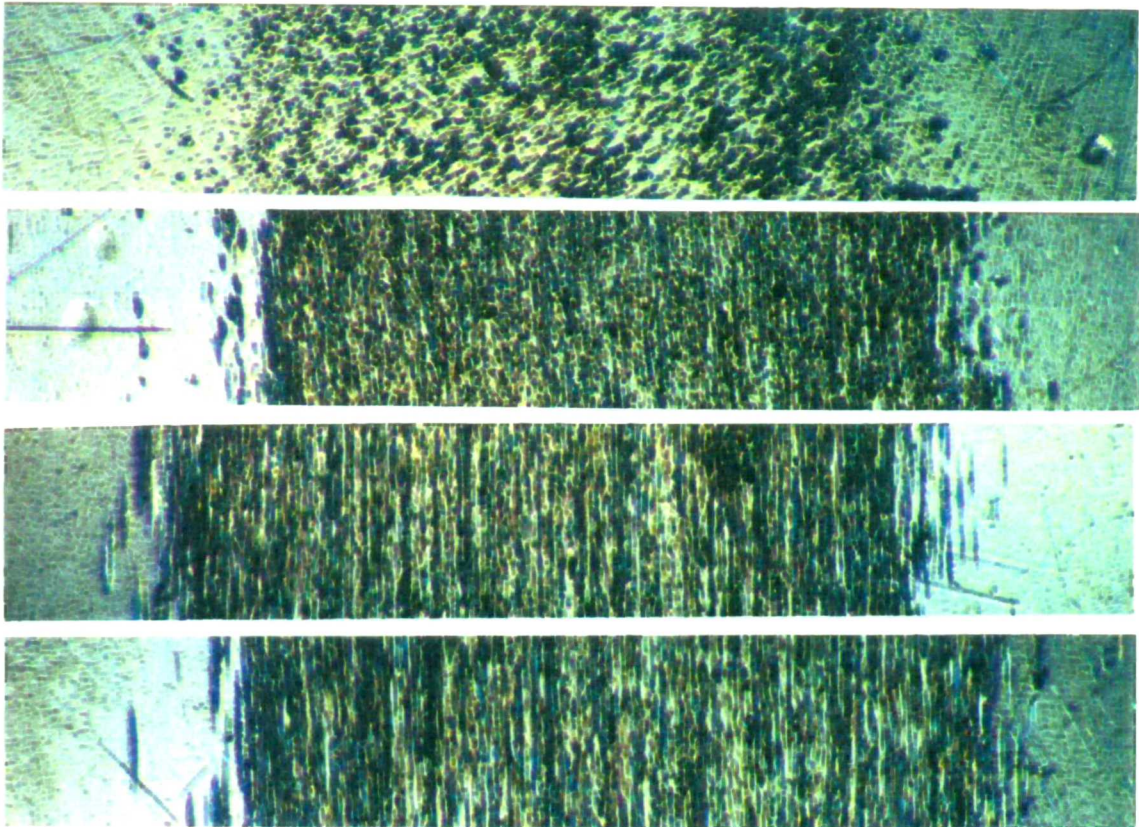


Figure 7.7. Photomicrographs of an abraded ball surface showing appearance of wear scratches for a range of slide roll ratios. Top to bottom 0%, 10%, 25%, 40% slide roll ratio. (Magnification x 100)

Figure 7.8 shows a Talysurf profile of a scratch in the direction of sliding (i.e. from the beginning to end of the groove). Unfortunately resolution is somewhat low, since the profile contains few data points because it is a sampled section of a larger profile;

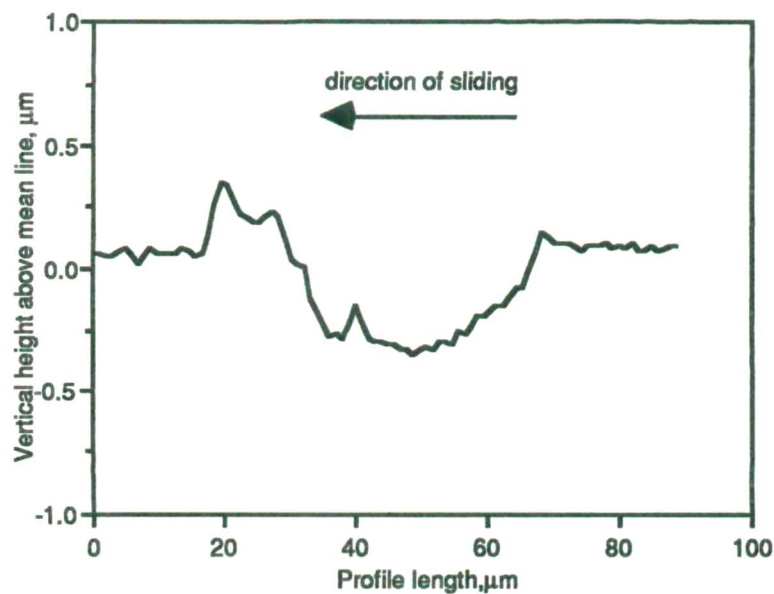


Figure 7.8. Surface profile of an abrasive groove (by 6 μ m silicon carbide particles) recorded in the direction of sliding. The direction of sliding is right to left.

The profile shows how material has been pushed ahead of the particle; the length of the groove is again consistent with the sliding distance (in this case $50\mu\text{m}$). Many of these formations appeared in the profiles recorded; however usually they are shorter than the sliding distance and less well defined, since the Talysurf stylus traverse does not always follow the full length of the groove, but only some section of it.

7.2.2.2. Abrasive Wear and Slide/Roll Ratio

The first series of tests showed how the wear grooves altered by varying the slide/roll ratio from 0% to -40%²⁵. In this case the slip on the ball surface was negative (i.e. in the opposite direction to rolling). Examination of the two contact surfaces showed surprising results. The surface of the disk specimen was marked by pits whilst the ball specimen was marked by grooves. All the relative sliding had been accommodated by the ploughing of the particle through the ball surface. For larger slide roll ratios the grooves on the ball became longer but the pits on the disk remained approximately the same size. Figure 7.9 diagrammatically represents these results.

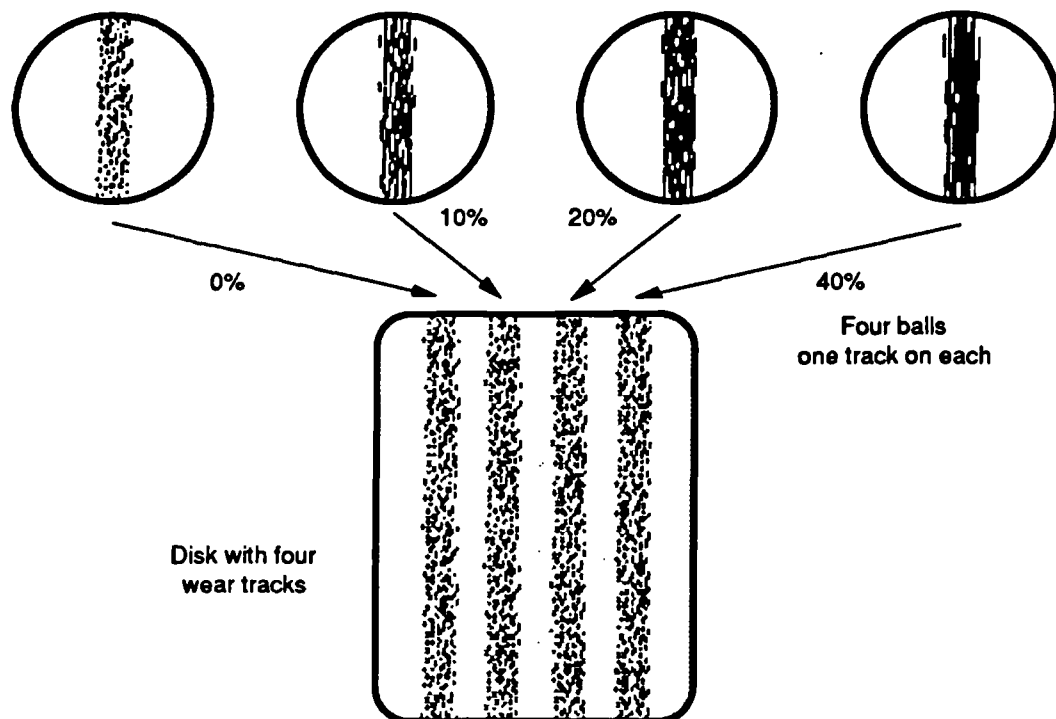


Figure 7.9. Schematic representation of the results from the first test series; variation in abrasive action with slide/roll ratio. Four ball specimens were run against different tracks on the disk with increasing slide roll ratio.

The lengths of the grooves was measured, they corresponded closely to a calculation of the expected sliding distance each particle would undergo (i.e. the slide/roll ratio multiplied by the contact width in the direction of motion). Figure 7.10 is a plot of the groove lengths for this and the next set of experiments. The data is compared with the predicted sliding distance.

²⁵ The slip is here defined as that at the ball surface. So a negative slip implies that the sliding direction is opposite to the rolling direction and, therefore, that the ball surface is slower. Slide roll ratio is defined by $2(u_b - u_d)/(u_b + u_d)$

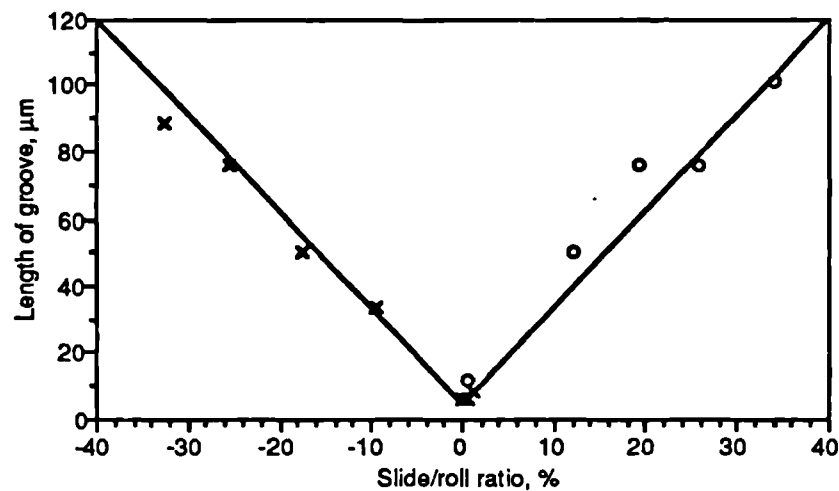


Figure 7.10. Plot of the length of the groove generated on the ball surface against the slide/roll ratio. Experimental measurement compared with a prediction of the sliding distance (calculated by multiplying the slide/roll ratio by the contact width in the direction of rolling).

It seemed surprising that all the grooving damage occurred on the ball surface. So the direction of sliding was reversed to see if this was the cause of the grooves being confined to one surface. The second series of tests were a repeat of the first but with a positive slide/roll ratio at the ball surface. The results were similar; the grooves being located on the balls whilst the disk remained only pitted. The results are also plotted on Figure 7.10.

The direction of sliding was clearly not responsible for this mechanism of preferential grooving. It was thought that the material properties may be responsible; another series of tests were performed to investigate the effect of hardness (and are discussed in §7.2.2.5).

The lengths of the grooves on the balls corresponded closely with the sliding distance. The dent depths on the ball however, varied little with slide/roll ratio. Table 7.1 shows the width and depth of the dents for the negative slide/roll series of tests.

Slide/roll ratio %	Ball Groove		
	Width, μm	Depth, μm	Length, μm
-40	2.8	.35	150
-33	3.1	.30	89
-26	3.6	.36	76
-17	3.0	.31	50
0	3.1	.22	6

Table 7.1. Variation in geometry of ball groove for the different slide/roll cases.

The dent depths are greater for the sliding cases than for the rolling case. In pure rolling the particle imbeds in the surface and after elastic recovery a residual dent is left. When sliding is present material is removed from the groove and thus after elastic recovery a deeper indent remains.

7.2.2.3. Abrasive Wear and Traction

The traction measurements from the first two series of tests (with some additional points under the same conditions) are plotted against slide/roll ratio in Figure 7.11. Some points from a test where no debris was present are included.

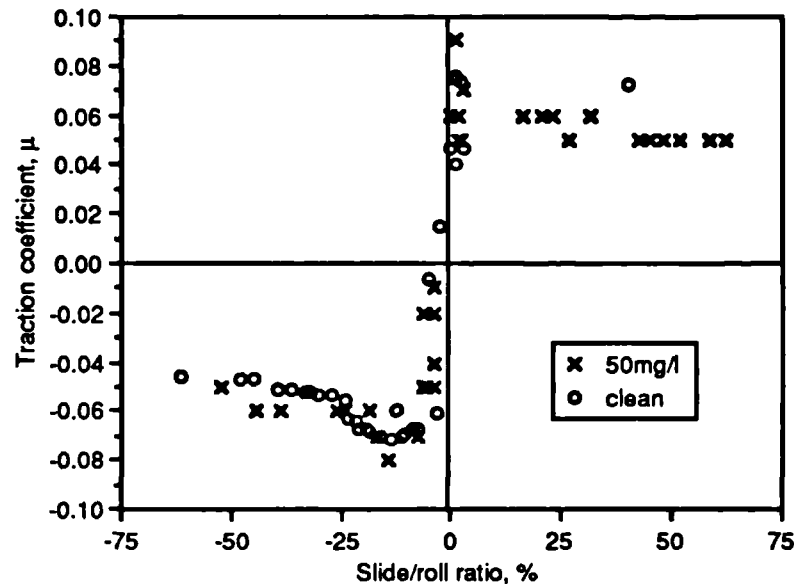


Figure 7.11. Traction coefficient variation with slide/roll ratio. Test data at a constant concentration of 50mg/l 6 μ m silicon carbide particles.

The traction coefficient was determined by the ratio of the traction force (measured by the torque tube) and the contact load. The plot shows the characteristic shape of a lubricant traction curve with a linear region at ratios less than about one or two percent sliding. At higher sliding the limiting shear stress is reached. This limiting value of the traction coefficient is approximately 0.05 to 0.06. This is in keeping with typical values of the limiting traction of other lubricants (in the range 0.04 to 0.06). The results for a clean lubricant are very close to those with contaminants (within the margin of error in the results, i.e. $\pm 10\%$).

Increasing the concentration of particles did not significantly affect this limiting traction coefficient. Figure 7.12 shows a plot of the limiting traction against debris concentration (for a slide/roll ratio of 12%). No trend outside the scatter of the data is visible.

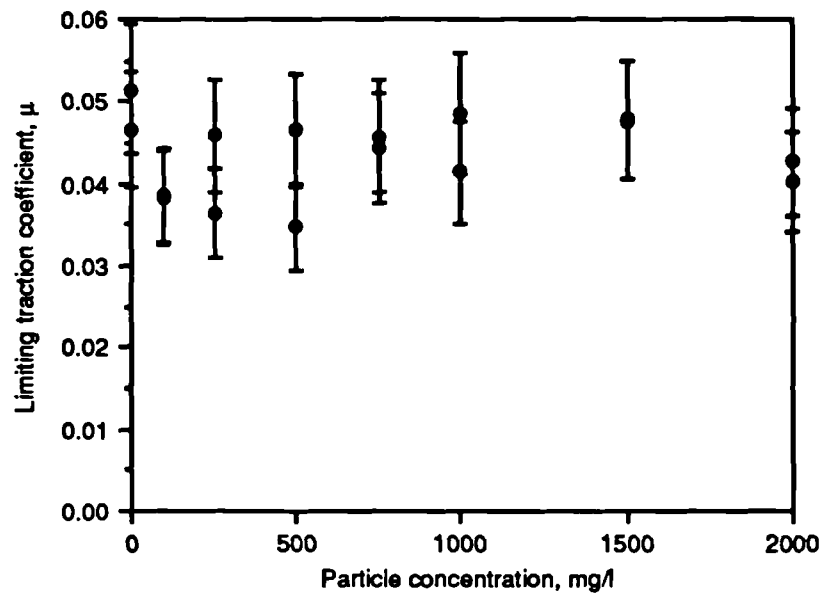


Figure 7.12. Traction coefficient variation with concentration of 6μm silicon carbide particles. Slide/roll ratio constant at 12%.

A simple calculation can be performed to predict the effect of particles trapped in the contact on the traction. Consider a single particle imbedded between the two surfaces and ploughing. The force on that particle, T_p , in the direction of motion, will be given by the hardness of the material, H multiplied by the projected ploughing area, A (assuming that the deformation is fully plastic and that the particle remains rigid);

$$T_p = AH \quad (7.1)$$

Then the friction coefficient is given by;

$$\mu = \frac{(T + nT_p)}{P} \quad (7.2)$$

where P is the applied load, n is the number of debris particles in the contact, and T is the traction force of the contact without debris. For 6μm particles imbedded in bearing steel with a film thickness of 0.6μm the ploughing force, T_p would be about 0.04N; and would make a contribution of approximately 0.1% to the overall traction coefficient. At the high concentrations used in the experiments, one might expect, at most, one hundred particles to be present in the contact. This would only contribute an increase of 10% to the traction coefficient. The measurement of these changes in traction were not possible with the existing equipment.

7.2.2.4. Abrasive Wear and Particle Size/Film Thickness Ratio

In the next series of tests, diamond abrasive powders of fixed size range were used (see Appendix 1). The lubricant conditions were fixed and the slide/roll ratio maintained at 20%. Table 7.2 shows the average dimensions of the abrasive scratches (sample size 15) for each of the size ranges used.

Size range, μm	Ball Groove	
	Depth, μm	Width, μm
0.0-0.5	-	-
0.5-1.0	.11	1.9
1.0-2.0	.15	2.1
2.0-4.0	.34	3.4
3.0-6.0	.9	4.9

Table 7.2. Average geometry of abrasion marks on the ball surface for increasing diamond debris particle size. Film thickness $0.44\mu\text{m}$, slide/roll ratio 20%.

Again all the grooves were generated on the ball surface whilst the disk was only pitted. The lengths of the grooves were close to the predicted sliding distance. For the smallest diamond size range the damage was not distinguishable from the surface roughness of the ball. The larger sizes cause deeper and wider dents. The width of the dents for the $0.5\text{--}1.0\mu\text{m}$ size band seems high. This is probably an anomalous feature of the profilometer. The stylus has a radius of $2\mu\text{m}$ and may not penetrate to the bottom of a sharp scratch (the minimum width measurable of a $0.1\mu\text{m}$ deep scratch is about $1.2\mu\text{m}$). Figure 7.13 shows photographs of the abraded ball specimens for each of these cases.

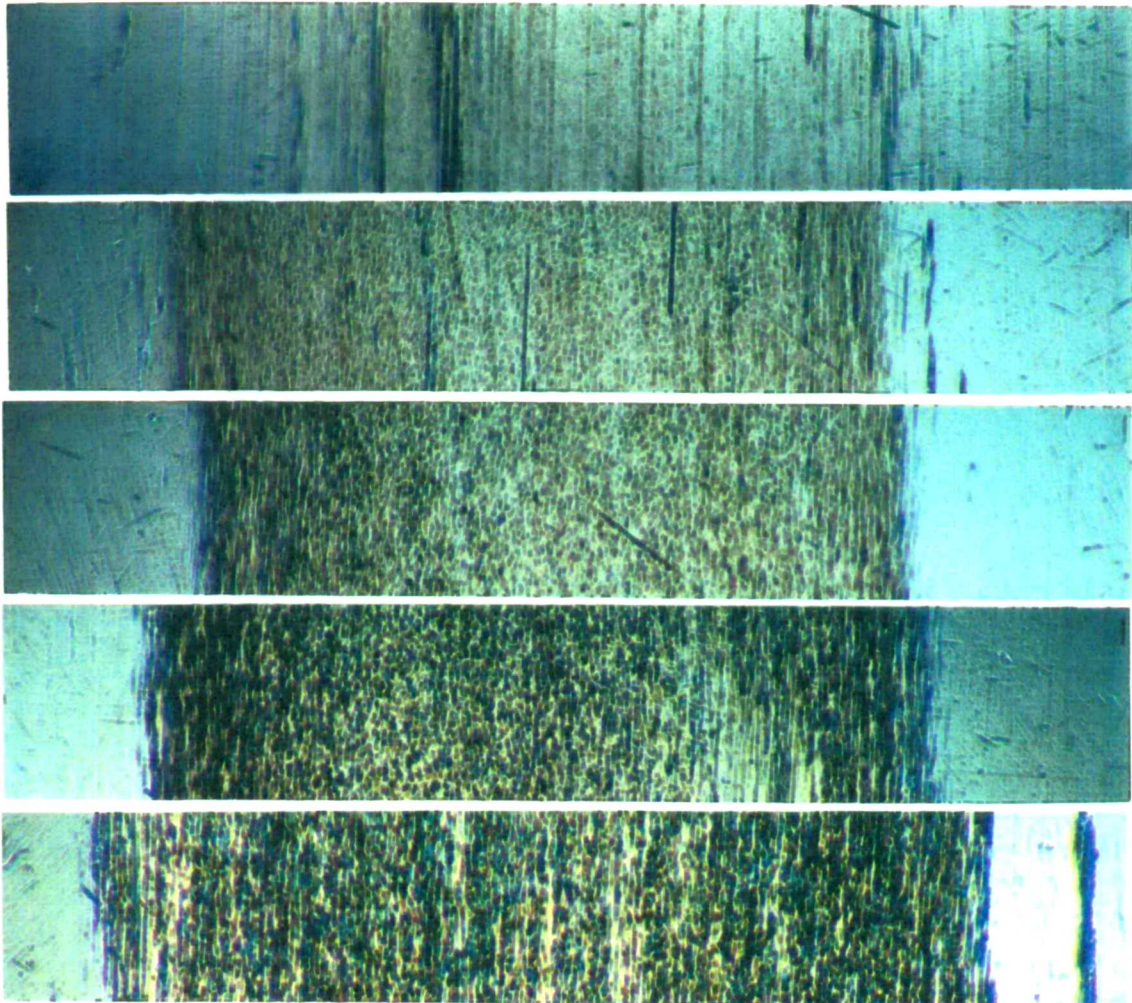


Figure 7.13. Photomicrograph of ball wear tracks from five tests using diamond abrasives of size bands $0\text{--}0.5\mu\text{m}$, $0.5\text{--}1\mu\text{m}$, $1\text{--}2\mu\text{m}$, $2\text{--}4\mu\text{m}$, and $3\text{--}6\mu\text{m}$. Film thickness $0.44\mu\text{m}$, slide/roll ratio 20%. (Magnification $\times 100$)

The ball tested with 0-0.5 μm diamond powder is hardly damaged. The few scratches visible may be from larger particles in the mix or from indentation caused during start-up or stopping of the rig when the fluid film may not have been generated.

The next two photograph strips show a very interesting phenomenon. The ball wear track shows distinct regions; a central region where mostly dents are present, and outer regions where the familiar grooves have been generated.

A different wear mechanism appears to be in operation. The reason for this becomes clear when one considers the nature of the ehl film. The film consists of a central region with a surrounding 'horseshoe' of lower film thickness. Hence the edges of the ball track are in a region of thinner film. Figure 7.14 schematically shows the ball track relative to its ehl film. The contact is a ball on cylinder and so the film footprint is elliptical.

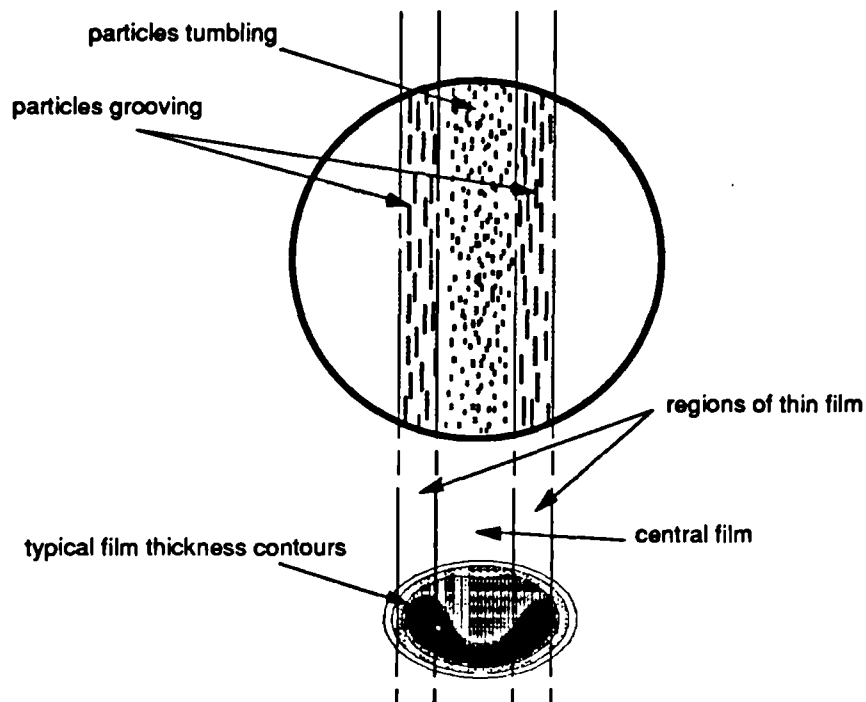


Figure 7.14. Schematic representation of the ball wear track from the test with 0.5-1.0 μm diamond abrasive powder alongside its ehl film.

The particles in the outer edges of the track are therefore being overrolled in a region of thinner film; more of their bulk is imbedded in the contact elements and the particles tolerate relative sliding motion by ploughing through the ball surface. In the central region less of the material is imbedded and the particles tend to tumble through the contact. Some grooves are present in the central region, this may be caused by larger particles in the debris mix, or because some of the overrolling is occurring in the thin film region at the outlet.

For larger diamond abrasive sizes (2-4 μm and 3-6 μm) the transition is no longer present. Ploughing is occurring across the whole contact width.

Williams (1992) has noted this ploughing and tumbling process using experiments on a foil bearing. He has developed a model to predict this change from a tumbling to a ploughing mechanism. He suggests that for tumbling to occur the ratio of particle size to lubricant film thickness must be less than about 2. At ratios greater than this value the mechanism will be ploughing.

Table 7.3 shows this ratio of particle size to film thickness (D/h) for each of the size bands for the thin and central film regions. It is clear that the particles in the central region for the 0.5-1.0 μm and 1.0-2.0 μm size band may be in this tumbling regime; whilst ploughing is likely at the thin film edges.

Size range, μm	Particle size/film thickness ratio, D/h	
	central film region	thin film region
0.0-0.5	0.0-1.1	0.0-1.5
0.5-1.0	1.1-2.3	1.5-3.0
1.0-2.0	2.3-4.5	3.0-6.1
2.0-4.0	4.5-9.1	6.1-12.1
3.0-6.0	6.8-13.6	9.1-18.2

Table 7.3. Ratio of particle size to lubricant film thickness for each size band of diamond abrasive used in the test series. Calculated²⁶ central lubricant film thickness 0.44.

It is true that more sliding occurs in these outer bands (microslip from the geometry of the contact, this is discussed further in §8.3.1). To check that this increase in sliding was not responsible for the above feature, the test was repeated at a higher 30% slide/roll ratio. The wear track was similar to that from the previous test (although with longer grooves in the outer region); no change in the width of these ploughing and tumbling regions was visible.

As a further check 1 μm alumina powder was also tested under the same lubrication conditions. The same mechanism of tumbling in the central regime and ploughing at the edges was observed. It is therefore suggested that this ploughing or tumbling mechanism is dependent on the particle size/film thickness ratio²⁷.

7.2.2.5. Abrasive Wear and Contact Element Hardness

It was suggested above that the reason for the preferential grooving of the ball was that the softer disk surface held the particles which then scratched the harder ball surface. To test this hypothesis three of the ball specimens were softened by tempering (see Appendix 1 for details). Again, they were run with 6 μm silicon carbide as an abrasive. Table 7.4 shows the hardness of the ball specimens against the resulting average scratch depth (for a sample size of 15 depths varied typically by 50% of the mean). The ball hardness of 839V is that for the untreated condition.

Ball hardness, VHN	Ball/disk hardness ratio	Dent depth, μm
839	1.1	0.32
672	0.88	0.66
578	0.76	0.72
434	0.57	0.85

Table 7.4. Ball specimen hardness against resulting scratch depth. Scratch depths are an average of 15 samples.

²⁶ Using Dowson & Higginson's (1977) equation.

²⁷ Although, recently Fang et al (1992) showed that particle shape can also control the tumbling or ploughing behaviour.

The dents are deeper on the softer ball surfaces. Measurement of the dents on the disk surface is difficult; since the surface roughness is of a similar order. In all the softened ball cases, grooves were now formed on the disk surface. Although grooves were still present on the ball surface. The mechanism of one surface holding the particle while it scratched the other did not swap completely with the change in hardness ratio. Various groove lengths were noted; it was thus not possible to predict particle motion on this basis.

It was expected that when the ball became the softer surface, it would imbed the debris and scratch the disk. This was only partially the case; grooves of various lengths appeared on both surfaces. The reason for this behaviour is not fully understood. Three possible reasons are; the rougher disk surface aiding the trapping of particles; the ball undergoing more cycles and work hardening; or the heat treatment process affecting the results in some way. A more rigorous test procedure is required to fully investigate this problem.

The test using the silicon nitride ball against the disk did not show groove formation on either surface (tests were performed at 10, 20, and 30% slide/roll ratio). Only very fine pits were left on the ball surface (about $0.1\mu\text{m}$ deep); the damage on the disk surface appeared as a single dent surrounded by a patch of smaller dents. The abrasive particle appeared to have been pushed fully into the disk surface and when sliding occurred the exposed top edge seemed to have fractured off. The relative sliding was therefore accommodated by particle fracture, rather than any ploughing action.

7.2.2.6. The Range of Abrasive Groove Sizes

The measurements of the abrasive grooves show a considerable amount of scatter. A large number (sample size 150) of scratches cut by $6\mu\text{m}$ silicon carbide were measured. The groove geometry data points have been plotted as a cumulative distribution on Figure 7.15 (the abscissa represents the percentage difference from the mean value).

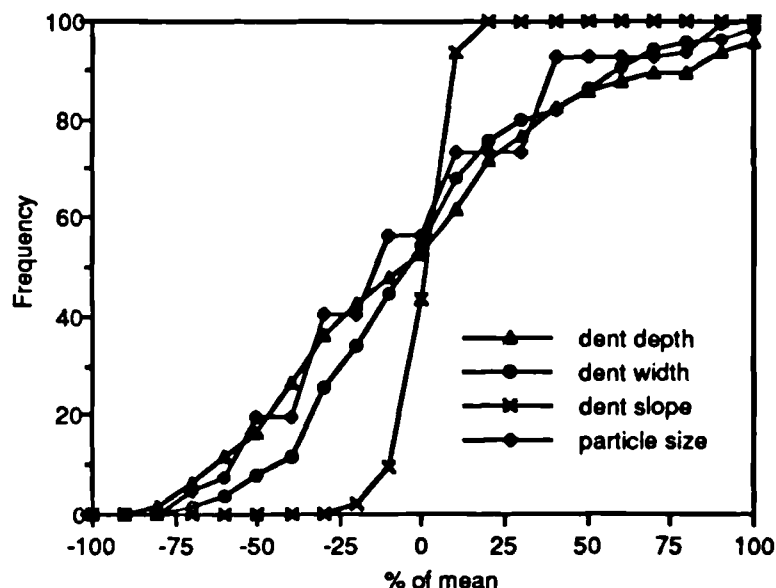


Figure 7.15. Cumulative frequency distributions of abrasive dent geometry, with abrasive particle size. The data has been normalised by displaying all values as a percentage difference from the mean.

The smaller grooves are barely distinguishable from the surface roughness, (0.2µm deep by 2.4µm wide), whilst the largest are as much as 1µm deep by 4µm wide. This wide range of groove sizes can be attributed to two sources; firstly the size range of the abrasive powder, and secondly the orientation of the particles. The three curves show the frequency distribution of the dent depths, dent slopes, and abrasive particle size, measured using the Malvern Size Analyser.

The first point to note is that the dent slope curve is steep and therefore the scatter in the data is low. So, even though the size of the dents vary, they are similar in shape. The trends of the dent depth, width and particle size data are similar. This suggests that the scatter in the data is also similar. Given these two pieces of evidence it is likely, therefore, that the scatter of groove size data is caused by the distribution of the abrasive particles. The abrasive grit shapes are similar, so a small abrasive grit will generate a dent of similar slope to a large one.

7.2.2.7. The Proportion of the Groove Removed as Wear

An important part of the study of abrasive wear is understanding how much material will be removed from the contacting elements by the action of an abrasive particle. This is clearly related to the size of the abrasive particle; although the volume displaced by the particle does not necessarily equal the volume of material removed. Some material may be elastically deformed or some displaced to form dent shoulders. So in general we can write;

$$A_{disp} = A_{elastic} + A_{plastic} + A_{chip} \quad (7.3)$$

where A_{disp} is the amount of material displaced by the abrasive particle, $A_{elastic}$ is the area of the indenter accommodated by elastic displacement of the surface, $A_{plastic}$ is the area accommodated by plastic displacement, and A_{chip} is the area removed as a wear chip. The plastic flow of material can be as the formation of dent shoulders or as the flow of material into the side walls.

The wear model of Buttery and Archard (1971) given in equation (6.6) includes a parameter β which is a measure of the proportion of material removed from the abrasive scratch. This parameter is determined from the shape of the abraded scratch. Their definition of β is given by the area of the groove less the area of the shoulders divided by the area of the groove (i.e. $(A_3 - A_2)/(A_3 + A_1)$ in Figure 7.16). This takes no account of the elastic accommodation of the abrasive particle or the amount plastically flowed into the dent side walls.

A more generally useful definition of the material removal parameter would be;

$$f = \frac{A_3 - A_2}{A_{disp}} \quad (7.4)$$

The area displaced by the abrasive grit is obtainable from the geometry of the abrasive grit. So in Figure 7.16;

$$A_{disp} = \frac{(\delta_1 - h)^2}{4} \frac{\delta_2}{\delta_1} \quad (7.5)$$

where h is the film thickness and δ_1 and δ_2 are characteristic dimensions of the abrasive particle.

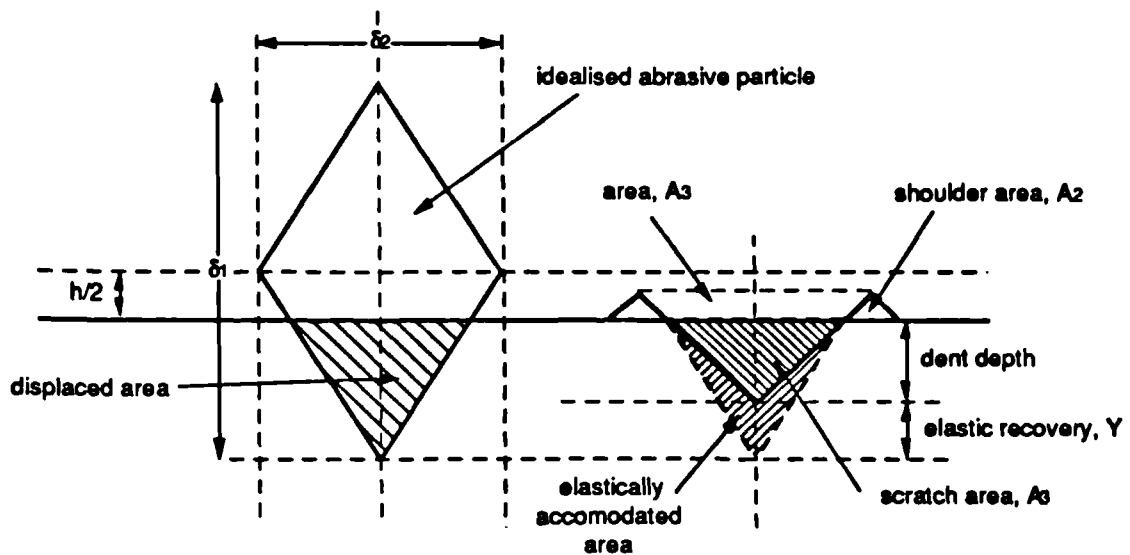


Figure 7.16. Diagram showing the area displaced by an idealised abrasive particle. Note also the definitions of Y , δ_1 , δ_2 , A_1 , A_2 , and A_3 .

So given this factor, f , and an abrasive particle geometry it is possible to predict the material removed from the surface. This is discussed in greater detail in §8.3.5, where the relevance of this factor to the modelling of abrasive wear is demonstrated.

It is relatively simple to determine this grooving factor from the experimental results. The area displaced by the particle is easy to calculate given an assumption about the particle geometry. The groove volume can be measured, and any built up material subtracted, from the surface profile results.

For the silicon carbide particles it was decided to choose $\delta_1 = 6\mu\text{m}$ and $\delta_2 = 4\mu\text{m}$ ²⁸ whilst the film thickness was $0.43\mu\text{m}$. This gives a displaced area of $5.17\mu\text{m}^2$. Table 7.5 displays the average values of the dent geometry parameters and the calculated factors.

Parameter	Average value
Dent depth	0.65
Dent shoulder height	0.08
Dent width	$5.5\mu\text{m}$
Dent slope	13°
Area, A_1	$0.53\mu\text{m}^2$
Area, A_2	$0.08\mu\text{m}^2$
Area, A_3	$2.17\mu\text{m}^2$
Area displaced, A_{dis} eqn. (7.5)	$5.17\mu\text{m}^2$
Removal factor, β eqn. (6.6)	0.77
Removal factor, f eqn. (7.4)	0.41

Table 7.5. Average values of scratch geometry, areas and calculated material removal parameters. Data from ball on disk tests using $6\mu\text{m}$ silicon carbide abrasive (sample size 150).

²⁸ These dimensions were chosen by careful inspection of the particles, using a microscope, and choosing the appropriate sizes.

The factor, f is somewhat higher than β predicted from the scratch measurements alone. The reason for this becomes clear when one considers the process by which the indenting particle is accommodated. The relation of Buttery and Archard (1971) only considers the plastic flow into the dent shoulders; it does not consider elastic accommodation of the particle or flow into the indent wall (this was noted by Johnson in his discussion to the paper and later by Torrance (1980a).

It is possible to estimate the area of the indenting particle which can be accommodated by elastic deformation of the surface. After the particle is removed elastic recovery takes place; the width of the indentation changes little but the depth reduces (Tabor 1951). Since the unloading process is essentially elastic (assuming no reverse yielding on unloading), then the recovery can be determined from the elastic displacement solutions. Such an analysis was performed by Stilwell and Tabor (1961) using the elastic solution of Love (1939). They give the elastic recovery of the depth as (see Figure 7.16);

$$Y = \frac{\pi p a}{E^*} \quad (7.6)$$

where a is the dent half-width (which does not change on unloading), p is the mean indentation pressure, and E^* is the reduced modulus. Since the indentation will be fully plastic the contact pressure is assumed to be equal to the hardness of the surface.

Using the average scratch data displayed in Table 7.5; the dent depth recovery, Y is predicted to be $0.5\mu\text{m}$ and so the elastically accommodated area is calculated as ($A_{\text{elastic}} = Ya$) $1.36\mu\text{m}^2$. If this is now included in the definition of β ;

$$\beta = \frac{A_3 - A_2}{A_3 + A_1 + A_{\text{elastic}}} \quad (7.7)$$

Then the corrected value of β is 0.51. This is now closer to the value of f , predicted from the assumed geometry of the abrasive shape. The error between the two probably now stems from the area of the indenter accommodated by plastic flow into the groove side walls.

The data from the series of tests using diamond abrasives was also used to determine this f factor. Again the area of the scratch was related to the cross sectional area of the abrasive particle (using equation 7.4). Table 7.6 shows the factors calculated for each diamond size band (the data from the smallest sized particles was thought to be unreliable for this prediction).

Size range, μm	Estimated f factor
1-2	0.35
2-4	0.28
3-6	0.51

Table 7.6. Estimate of the f factor predicted from the size of grooves measured for the tests with a range of diamond abrasive particles.

The data is of the same order as with the $6\mu\text{m}$ silicon carbide data. But, it is difficult to discern any further trend given the shortage of data. The scratch measurement process is difficult and may lead to the variation in f factor with particle size noted here. Further work is required to investigate this fully.

7.2.2.8. Abrasive Wear and Debris Materials

The final series of tests were a brief investigation into the behaviour of other ductile debris materials. Copper and low carbon steel particles (all sized 58-63 μm) were used as a test abrasive. The concentration was maintained at 25mg/l and the slide roll ratio at 20%.

Both debris material caused the formation of grooves on the ball surface and only dents on the disk. Table 7.7 shows the geometry of the grooves for each material.

Abrasive material	Groove depth, μm	Groove width, μm
low carbon steel	0.59	70
copper	0.47	8

Table 7.7. Average geometry of grooves formed on the ball surface by the abrasive action of copper and low carbon steel particles.

The dents are shallower and narrower than those found in the pure rolling studies (see Table 3.3). They are still considerably larger than those created by the ceramic debris materials. One might expect the ductile particle to shear when the two surfaces undergo relative sliding. Instead, the relatively soft particle is causing the formation of a groove on a hard surface. When the particle first becomes squashed interfacial friction causes the contact pressures to exceed the yield stress of the steel surfaces (see §3.4.1) and a dent is formed. If the two element undergo relative sliding then these high pressures are still likely to be present. They will cause rolling element yield at the new particle position. This is perhaps how the grooves are formed.

7.3. The Bearing Tests

The ball on disk experiments provided interesting data on the three body wear process. The next set of tests was performed to provide data more specific to the rolling bearing abrasive wear problem. Some degree of idealisation has been incorporated into the experiment design. A compromise has been drawn, between providing useful practical data, and results which reveal more about the wear process.

Again small size ceramic debris particles were used (to remove the effects of the complex particle fracture or deformation). Loads, concentrations, and debris materials were also varied. The results were related to the fundamental abrasion mechanisms described §7.2.

7.3.1. The Bearing Test Rig

The test rig consisted of two deep groove ball bearings mounted on a shaft either side of a large loading deep groove ball bearing. The two small bearings act as test specimens whilst the large central bearing transmits the load. The bearings were press fitted onto the shaft and held in place by nuts. The outer races were mounted in the housing. The housing was completely enclosed, with penetrations for lubricant and the shaft coupling. A hydraulic loading stud was used to apply load to the central bearing. This central bearing was entirely supported on the shaft so the load was therefore transmitted directly to the test bearings. Figure 7.17 shows the shaft and bearing assembly with the load and support points. The photograph included as Figure 7.19 shows the same assembly.

Lubricant was stored in a conical sump below the test chamber and was pumped through jets impinging directly onto the test and load bearings. Tappings in the lubricant feed were available

to record oil temperature and supply pressure. The pump intake was at the cone apex to ensure the entrainment of dense particles. The lubricant in the sump was cooled by cold water fed through a copper coil.

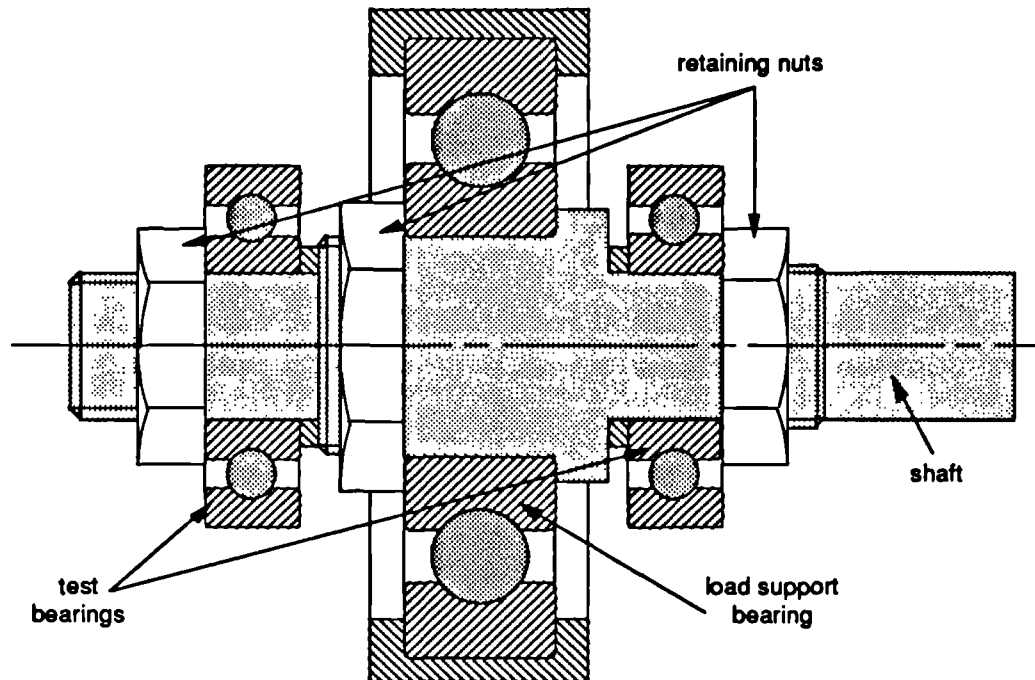


Figure 7.17. Schematic of the main shaft from bearing test apparatus.

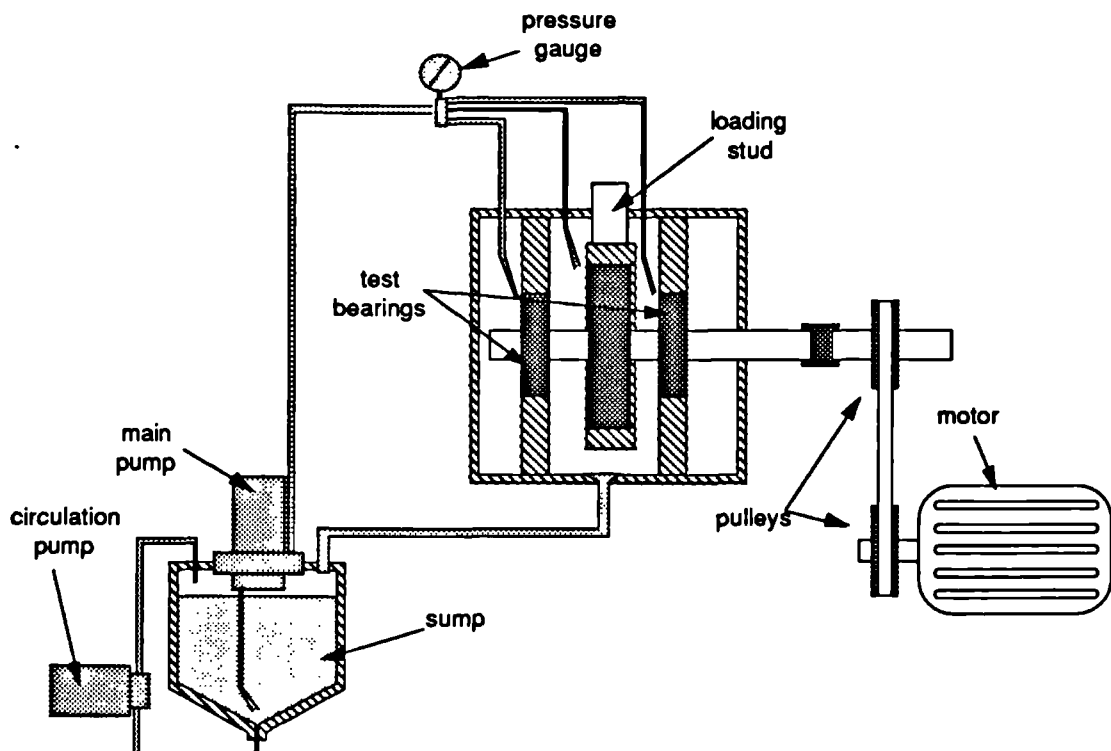


Figure 7.18 Schematic of the bearing abrasive wear test apparatus.

The shaft and bearing assembly were connected to a belt and pulley through a flexible coupling. A three phase electric motor rotated the shaft at constant speed. Figure 7.18 shows a schematic of the whole test apparatus. The photograph included as Figure 7.20 shows the test head.

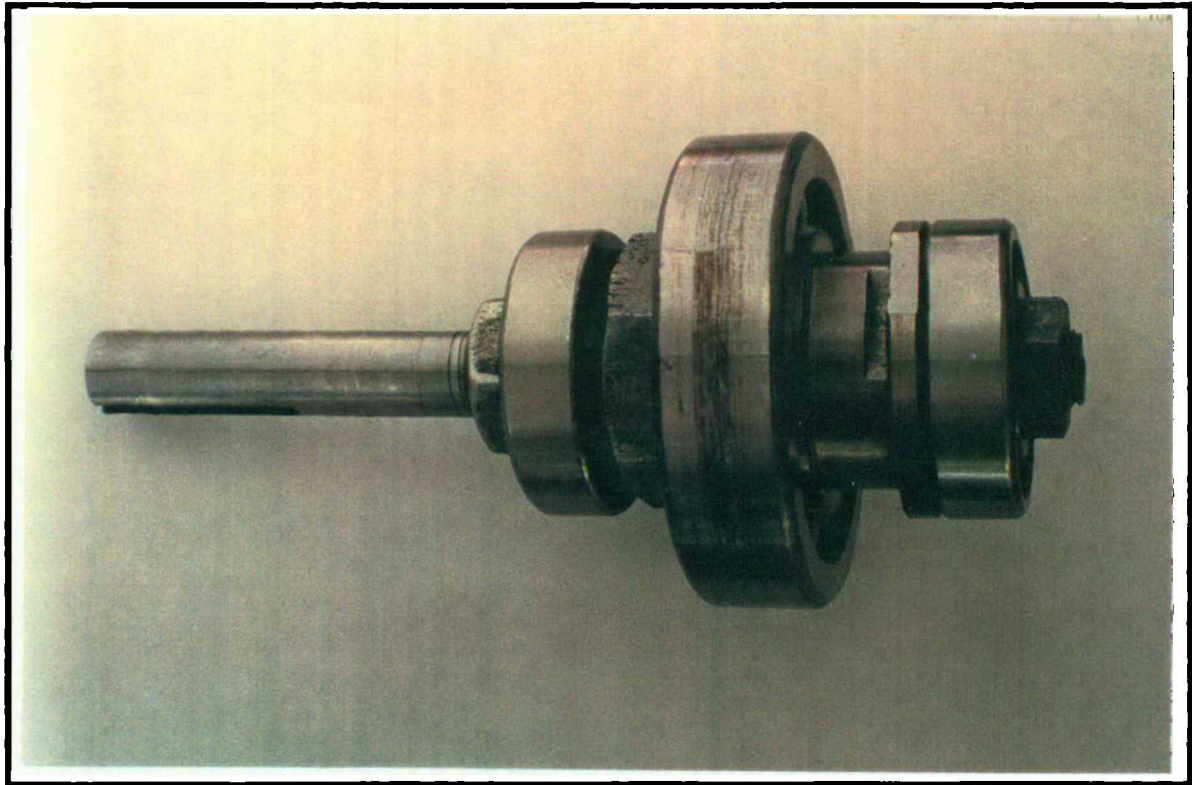


Figure 7.19. Photograph of the main shaft from bearing test apparatus.

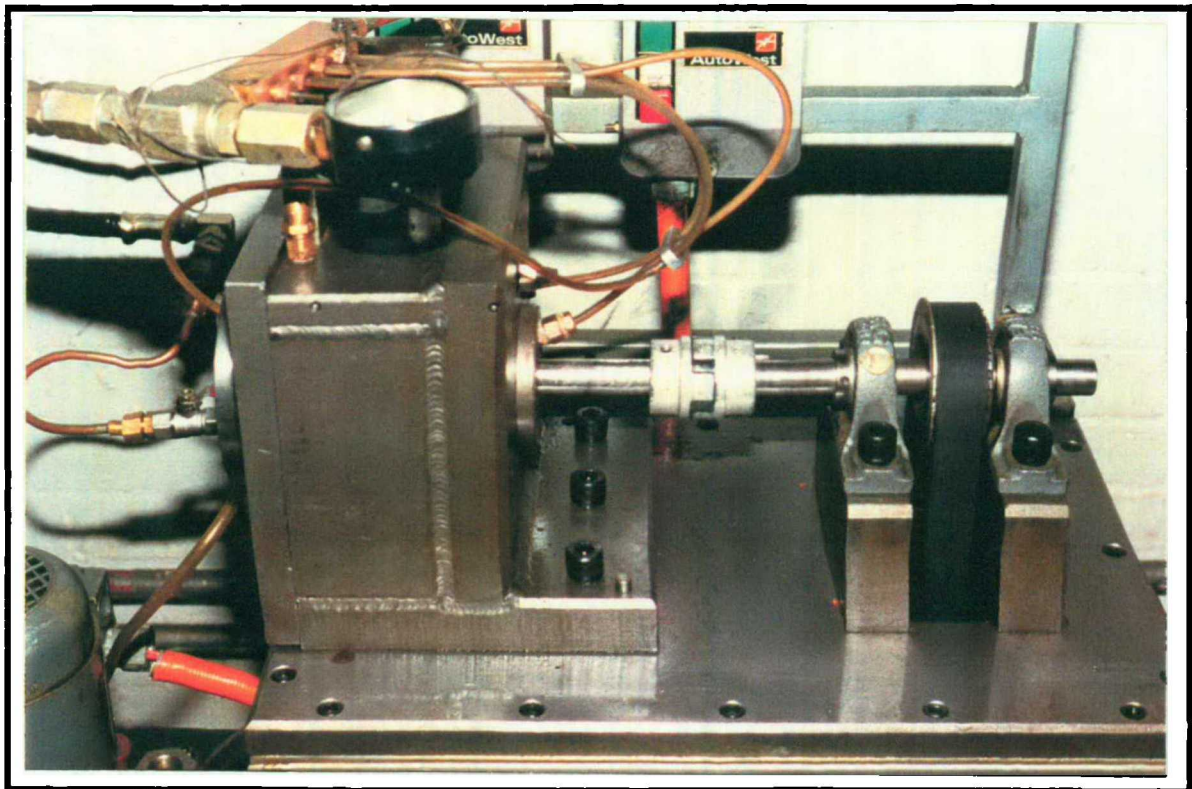


Figure 7.20 Photograph of the bearing test apparatus head.

7.3.1.1. Test Specimens

The bearings under used were standard deep groove ball bearings with plastic cages. The cages could be removed in order to disassemble the bearing for measurement of the mass of the individual components. The central support bearing, which had to be replaced frequently since it also became worn, was a larger deep groove ball bearing.

The lubricant used was Shell Turbo T68, with contaminating particles of 6 μ m silicon carbide and Arizona test dust (see Appendix 1 for a full description).

7.3.1.2. The Test Programme

The test scheme consisted of six series of tests designed to study as many of the relevant parameters as possible.

- (i) **Initial Test.** The first test carried out was performed to gauge the levels of abrasive wear to be expected from a given debris concentration. These results were then used to develop test schemes and time scales for the main bank of experiments. A relatively large concentration of 6 μ m silicon carbide debris was added and the test continued until excessive vibration was noted. Wear volumes and test durations were noted.
- (ii) **Effect of Debris Concentration.** The next tests were controlled more closely. Firstly a lower concentration of 6 μ m silicon carbide debris was used (1g/l); the test duration was eight hours with interruptions at two hourly intervals to sample the lubricant and weigh and measure the bearings. The test was then repeated for a still lower concentration (0.25g/l).
- (iii) **Effect of Debris Material.** In this set of tests Arizona dust was used as an abrasive at a concentration of 1g/l. The test duration was eight hours with two hourly measurement intervals.
- (iv) **Ball Wear with Respect to Cup Position.** The cage in a rolling element bearing is generally not machined to the same high tolerance as the elements and raceways. It was thought that slight irregularities in the ball cage contact might dramatically effect ball wear. In this test the balls were kept returned to the same cage pocket at each measuring interval. The balls were weighed individually and the mass loss determined.
- (v) **Effect of Bearing Load.** The tests using silicon carbide and Arizona dust were repeated at half the original bearing load; all other conditions remaining the same.
- (vi) **Statistical Repetitions.** To establish the repeatability of the results a series of ten test was performed under exactly the same contamination conditions; an uncontaminated control case was included. Extra care was taken to assembly the rig in the same manner (it was thought that bearing and shaft mounting may be a source of error) for each test. The tests were of duration eight hours with no interruptions for measurement.

7.3.1.3. Experimental Method

The test procedure in each of the experiments was basically the same. Six litres of fresh oil was introduced into the sump and circulated through a 5 μ m fine cartridge filter to remove any initial contamination. Debris powders were weighed accurately and mixed with a small quantity of oil

to make up a paste. This paste was gradually diluted and added to the bulk lubricant in the sump.

The apparatus was run for a few minutes under no load to circulate the debris throughout the oil and pipe work. It was essential to keep the debris in suspension throughout the test, the particles readily drop out of the lubricant and settle in the sump. Continual circulation of the oil, from the sump base to fluid level, was maintained. Load was then applied to the assembly and the test commenced. The tests were normally of eight hours duration; at two hourly intervals the test was interrupted for measurements to be made.

At the end of the test the oil was drained and the apparatus cleaned thoroughly and flushed with clean oil.

7.3.1.4. Test Measurements

Prior to commencement of the test and at two hourly intervals throughout the test, the following measurements were made;

- (i) **Mass Loss.** The individual components (balls, inner race, outer race, and cage) of the test bearings were weighed before and at intervals during the test. The plastic caged bearings allowed disassembly. The mass losses were generally very small, of the order of a few milligrams; it was, therefore necessary to thoroughly clean the components before weighing. An ultrasonic toluene bath and oven drying were used. An electronic balance accurate to ± 0.1 mg was employed.
- (ii) **Change in Surface Profile.** The change in surface profile of the steel components was measured during the tests. Profiling was carried out using a Form Talysurf. The data was fitted to a measured radius (this was maintained constant for each component throughout the test) to remove the curvature of the specimen. Roughness parameters were also recorded.
- (iii) **Bearing Clearance.** The bearing clearance was measured by re-assembling the bearing and mounting it under the Form Talysurf with the stylus on the inner bore. During the traverse the inner race was lifted whilst holding the outer race clamped still. In this way, the stylus followed a track including a step jump measuring the bearing clearance. This rather crude method gave surprisingly good, repeatable results.

7.3.2. Results

It is convenient to discuss the results of these tests in terms of the parameters involved, rather than by test sequence. A clearer picture of the abrasive wear process can be built up in this way. Since scatter is evident in the experiments, the results are presented as averages of the available data at the required test conditions.

7.3.2.1. Breakdown of Mass Loss in a Bearing

The measurement of mass loss for the individual components within the bearing allowed an investigation into how the total mass loss was apportioned between the bearing elements. This distribution of material loss between the bearing components provided interesting information about the possible wear mechanism.

Two abrasive material cases are described; 6 μ m silicon carbide, and Arizona dust which was sieved to give a size range of 58 to 63 μ m.

7.3.2.1.1. Silicon Carbide Abrasive and Ploughing

The pie chart included as Figure 7.21 shows the average breakdown of mass loss in a bearing tested with 6 μ m silicon carbide abrasive. The sample size was 40 (i.e. ten tests with four measurement intervals each). The calculated percentage breakdowns were 82 % of the mass loss occurring from the balls, 10% and 4% from the inner and outer races, and 4% from the cage. The data varied by typically 30-50% from the mean.

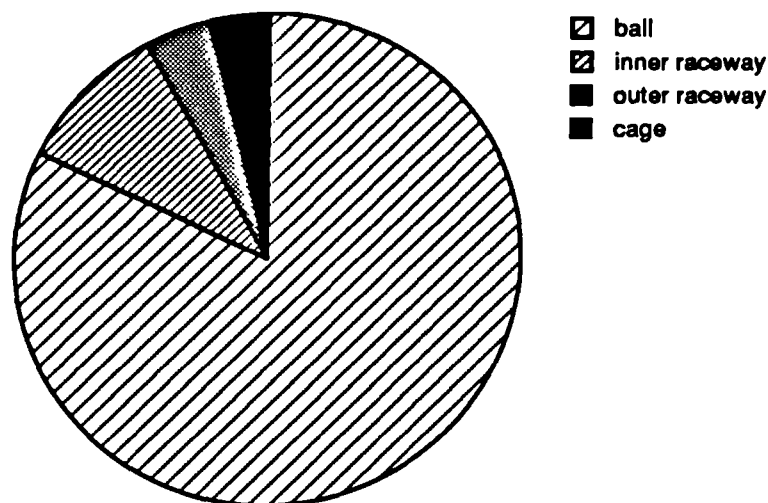


Figure 7.21. Pie chart illustrating the breakdown of mass loss between the components of a ball bearing (balls, inner race, outer race, and cage) tested with 6 μ m silicon carbide.

When one considers the kinematics of a ball bearing and the mechanism of abrasive wear, it seems unusual that so much mass loss occurs from the balls. The contacts, and therefore possible wear sites, within the ball bearing are;

- (a) Ball on inner race
- (b) Ball on outer race
- (c) Ball on cage
- (d) Erosive wear of all components (particularly those moving)

The cage contact may be responsible for some of this ball wear since it is fully sliding. However the cage surface seemed little damaged by the wear process, little imbedded debris was found, which could repeatedly score the ball. To remove, or at least alter, the effects of the cage, a steel caged ball bearing was tested. Since it was not possible to weigh the individual components before the test, estimates of the breakdown had to be made. A similar pattern emerged, most wear occurring on the balls, less on the races, and a slight burnishing of the cage.

Erosive wear is an unlikely source of mass loss at these low fluid velocities and short test durations.

It might be expected, at first sight, that the sum of the inner and outer race wear would equate, approximately, with the ball wear. This is far from the case, it is clear that some preferential wear mechanism between the ball and race contacts is occurring. The ball on disk experiments described in §7.2 have helped explain the mechanism. In the contact between the softer race and the harder ball all the relative sliding was taken up by groove formation on the ball. This same process is probably occurring in the rolling bearing leading to higher ball wear.

The bearing components were examined using an optical microscope to see whether grooves were formed on the balls and pits on the races. However the sliding distances in the contacts are small (ranging from about 2 to 16 μm) and it was difficult to conclusively distinguish between the two formations.

The profiles described in §7.3.2.2 below (and pictured in Figure 7.26), show that some of this relative sliding must have been accommodated at the raceway surface. The raceway profiles show higher wear in the regions of maximum Heathcote slip. It is suggested, therefore, that preferential ball wear occurs according to some relation which defines the proportion of relative sliding occurring at each surface. This relation is likely to be dependent, in some way, on the relative hardness of the two rolling element surfaces. In the abrasive wear model this feature was expressed by inclusion of the parameter g (see §8.3.4).

7.3.2.1.2. Arizona Dust Abrasive and Tumbling

The pie chart included as Figure 7.22 shows the average breakdown of mass loss in a bearing tested with Arizona test dust. The breakdown is somewhat different to the previous case; now 50 % of the mass loss occurred from the balls, 28% and 12% from the inner and outer races, and 10% from the cage. The sample size was 14 and the data varied by typically 40-60% from the mean.

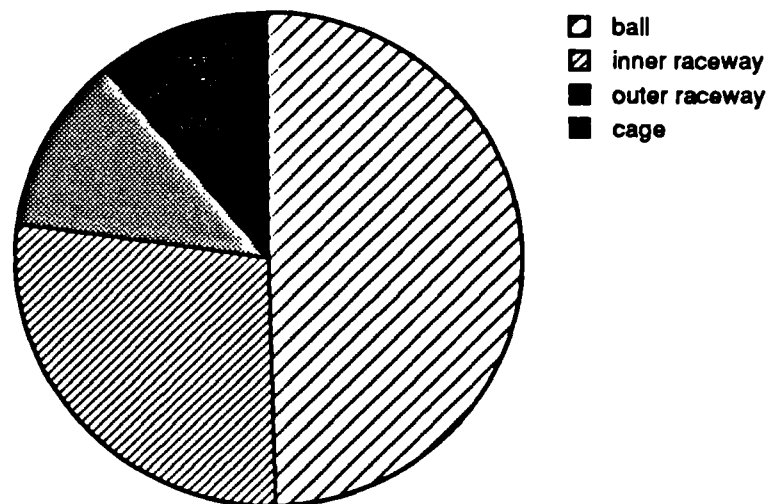


Figure 7.22. Pie chart illustrating the breakdown of mass loss between the components of a ball bearing (balls, inner race, outer race, and cage) tested with Arizona dust.

With this contaminant the wear is much more evenly distributed between the rolling elements. The sum of the inner raceway wear masses is close to that of the balls. The mechanism of preferential grooving, controlled by the element hardness, and found with silicon carbide, seems now no longer apparent.

It is possible that the wear mechanism between ball and race is no longer a ploughing process but a tumbling process. The Arizona dust particles fracture in the entry regions to the ball bearing contacts. The value of the fragment particle size to film thickness ratio, δ/h is approximately 1.7 (as oppose to 3.8 for silicon carbide tests); the critical value for the transition from a tumbling to a ploughing mode is thought to be about 2 (see §7.2.2.4). The ball on disk experiments showed that when tumbling occurred the damage on the two elements was similar (when ploughing occurred the damage was very different, with grooves on one specimen and pits on the other). One would thus expect that in a tumbling, as oppose to a ploughing mechanism, the mass loss would be significantly less and the distribution throughout the bearing would be more even. These features are noted in these Arizona dust tests when compared with those using silicon carbide.

7.3.2.2. Worn Profile and Appearance of Bearing Surfaces

The balls and races worn by abrasive particles have a frosted appearance, even the specimens from short duration tests have lost their reflective finish. A circumferential wear track, of uniform width, is visible on the inner race groove. A track on the outer race is clearly visible, but tapers from the loaded to the unloaded region. These 'tracking bands' are normally visible on the raceways of worn bearings. Figure 7.23 shows a close-up of the tracking bands on an inner raceway from bearing tested with Arizona dust.

Tracks are also visible on the balls (but only from tests which have not be interrupted and the bearing reassembled and rerun). The worn pattern indicates that the balls generally process about only one axis, parallel to the axis of the bearing. The balls which have been removed and replaced at each test interval show an evenly worn surface.

Both the metal and plastic cages show a burnished surface. Figure 7.24 shows a photograph of the wear marks on a plastic cage and a pressed steel cage. The pockets on the plastic cage have triangular wear regions at the sides, the worn area is larger on the driven side of the cage (since the loads and therefore contact areas are higher). The steel cage shows a similar wear pattern with two regions of abrasion where the ball is held. In both cases the radius of the cage pocket is less than that of the ball, so the ball contact tends to be in regions at the edge of the pocket (see Figure 7.25 for a visualisation of these wear patterns).

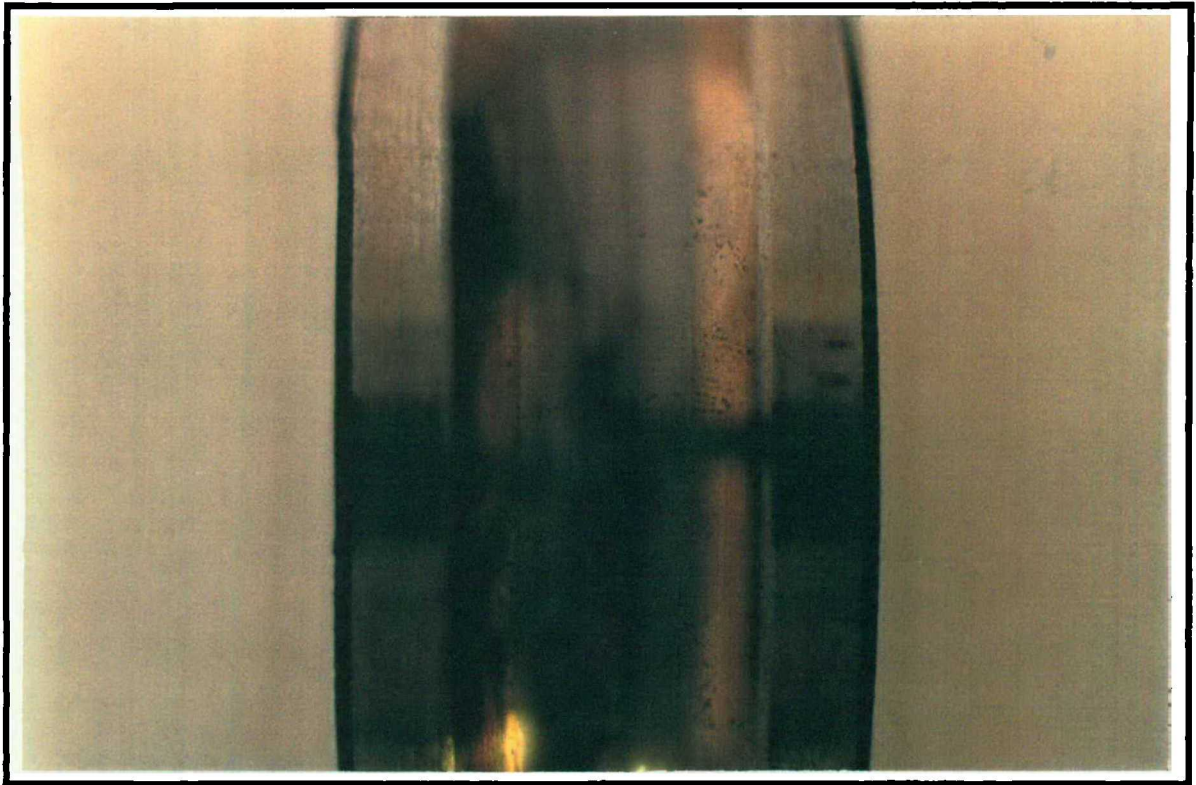


Figure 7.23. Photograph of tracking bands on an inner raceway from a bearing tested with Arizona dust abrasive.



Figure 7.24. Photograph of wear marks on a plastic and steel cages from bearings run with $6\mu\text{m}$ silicon carbide abrasive.

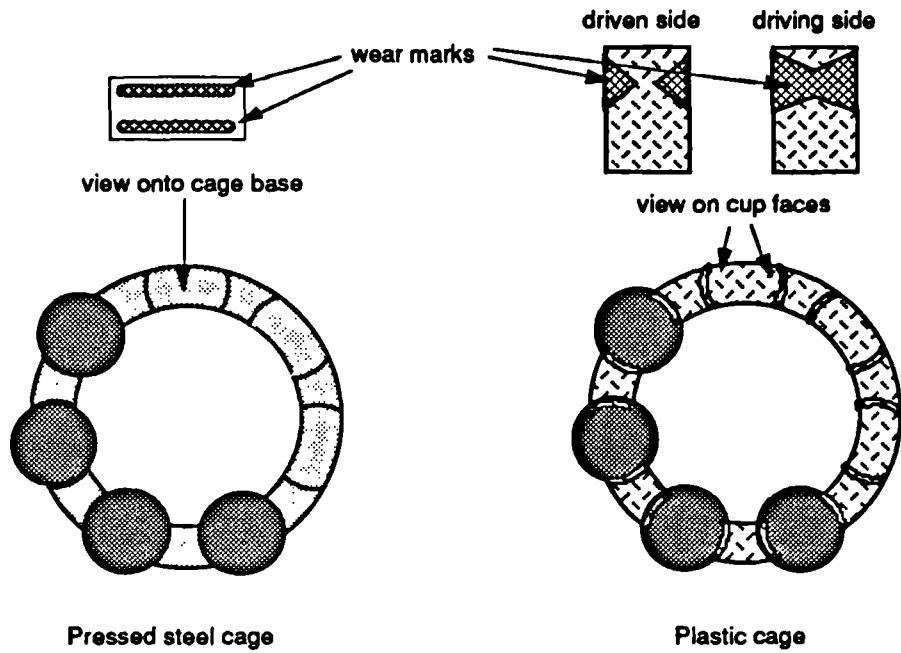


Figure 7.25. Diagram of wear marks on (a) steel and (b) plastic cages and sketch of the balls in a cage pockets. Notice wear on the leading side of the plastic cage cup face is greater than that on the trailing side.

Figure 7.26 shows two surface profiles from the groove of the inner race of a bearing tested at SKF ERC using Arizona dust as a lubricant contaminant. The profile is transverse to the rolling direction and follows from one side of the race to the other.

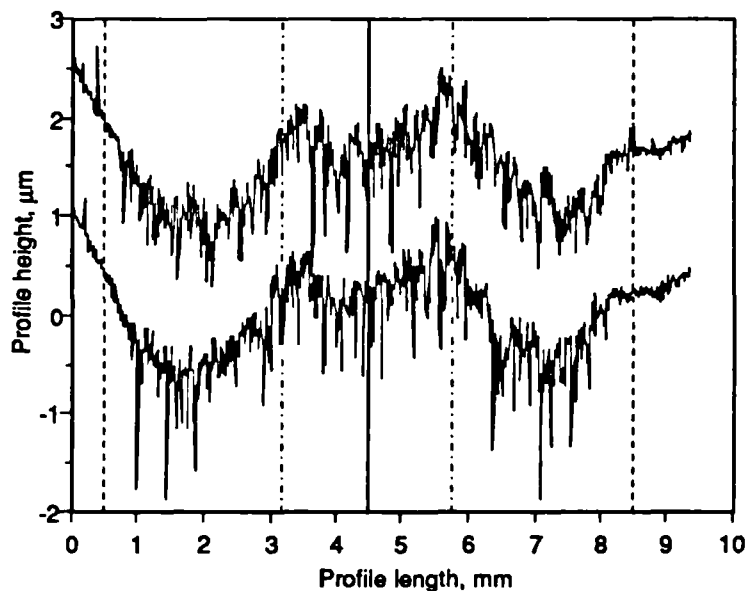


Figure 7.26. Two surface profiles, transverse to the rolling direction, of an inner race worn by Arizona dust. The two surface profiles are from sections 180° apart on the ring. A convex radius has been fitted to the data to remove the curvature of the inner raceway groove

The vertical dotted lines show the extent of the contact patch²⁹ between the ball and the raceway. The profiles show three troughs; the central trough being slightly less deep. Two peaks are also noticeable. These regions of light wear correspond to the regions of pure rolling within the contact. The positions of these pure rolling lines has been calculated (see §8.3.1) and are shown as vertical chain dotted lines. The troughs correspond to the regions of maximum microslip (or Heathcote slip).

The two profiles are from different positions on the inner raceway; both show very similar patterns. This process of transverse profiling at various positions around the raceway circumference was repeated for the outer ring. In this instance the profiles are not the same. Figure 7.27 shows a series of three profiles from around the outer race circumference.

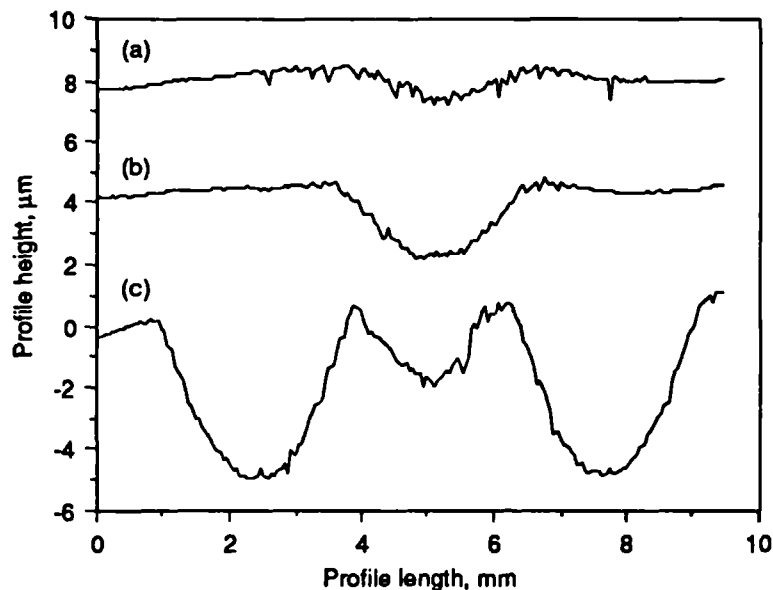


Figure 7.27. Surface profiles, transverse to the rolling direction, of an outer race worn by Arizona dust. The profiles corresponding to three positions around the circumference; (a) the unloaded region, (c) the loaded region, and (b) in between.

In the loaded region of the outer race the profile is similar to that of the inner race, with a pattern of troughs and peaks corresponding to Heathcote slip regions. In the unloaded region, a wear track still exists but the troughs and peaks are not so evident. This wear track is caused by the centrifugal loading of the balls onto the outer race.

This form of this wear profile is further evidence that wear is being controlled by the final fracture fragments when debris particles are broken down. The profile follows the regions of microslip; these are confined to the contact where wear is performed by the fracture fragments. If wear were to be initiated by large particles it would take place in the inlet region, where the regions of slip and no slip are not present. This would produce a uniform depth wear track and not the peaks and three troughs which are seen here. It is possible that some wear occurs in this way, but it is clear from the wear profile that the majority is controlled by the small fragments in the contact.

The above measurements were taken from a bearing tested at SKF ERC. A similar analysis was attempted using the test specimens from these experiments. At each of the test interruptions a

²⁹ As calculated using the Hertzian elastic contact equations.

profile was made of the inner race, the same radius was fitted to each profile to maintain a comparison. Figure 7.28 shows five traces for the inner race profile prior to test, and after two, four, six, and eight hours.

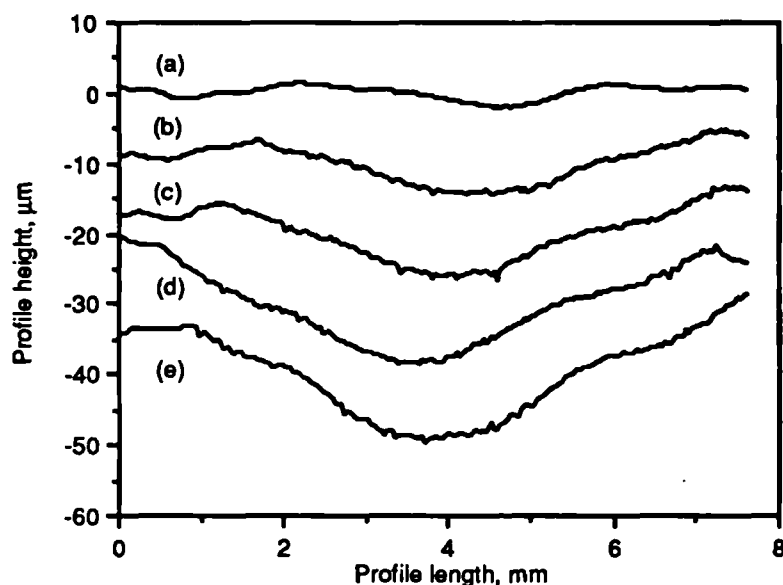


Figure 7.28. Surface profiles, transverse to the rolling direction, of an inner race worn by $6\mu\text{m}$ silicon carbide. Five profiles corresponding to ; (a) prior to test, (b) after two hours, (c) after four hours, (d) after six hours, and (e) after eight hours

The profiles do not show as clearly the development of troughs in regions of high slip. A deep trough has formed in the middle of the trace with two shallower ones at the sides. The bearing rig is not as well aligned as those used at SKF ERC. It is likely that some misalignment has allowed a degree of lateral motion of the balls; this has obscured the three trough formation.

7.3.2.3. Statistical Repeatability

The measurement of small relative mass losses is difficult and errors are easily introduced. To assess the repeatability of the data, five experiments were performed under exactly the same conditions using Arizona dust; a control was also run without contaminant. Two bearings were tested in each experiment, this gives a total of ten test specimens. Figure 7.29 shows the results in the form of a stacked bar chart (each bar includes a segment for the wear from each of the four bearing components). The data for the ten specimens is plotted alongside the results from a similar test using silicon carbide particles; the control bearing showed no mass loss.

The repeatability of the results is somewhat disappointing; the results vary by as much as 50% from the mean. However when compared with different test conditions (using silicon carbide abrasive) significant differences are evident. It is difficult to assess whether one particular feature of the results is responsible for the variation within a given test condition. The mass losses for each of the components showed a high variability. In addition, variation occurred between the two bearings in a single test; this implies that the lubricant supply conditions were not responsible for the variation.

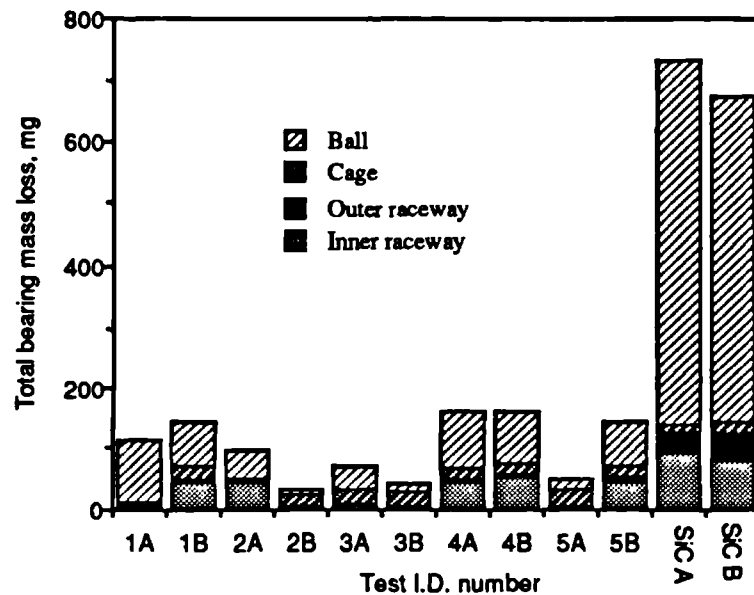


Figure 7.29. Mass loss for ten bearings tested with Arizona dust, and two tested with silicon carbide. (Concentration 1g/l, bearing load 4500N).

It was thought that the direction of the lubricant jet in relation to the bearing may play a part (the cage has one face which is slightly more open than the other). One test (number 5) was repeated altering the cage location; however, no correlation could be made.

The rig has no facility for adjustment of the test bearing alignment. The bearings are push fitted onto a shaft and into the housing. It is possible that a slight misalignment in the shaft assembly during assembly of the rig could be responsible for the mass loss variability. It is unlikely that other forms of, more statistically variable, wear are occurring (such as adhesive or chemical wear). The control test, performed using a clean lubricant, showed no mass loss, within the accuracy of the measuring apparatus (± 0.1 mg) and cleaning procedure (± 1 mg).

7.3.2.4. The Cumulative Wear Process

The graph included as Figure 7.30 shows all the bearing wear data plotted as a fraction of the total wear mass against the number of cycles. The data are for different test conditions; but have been normalised by dividing the wear at each measurement stage by the final wear mass. The average of the data is shown as a line on the plot.

The average mass loss is clearly linear with the number of cycles; however some scatter is evident in the individual tests. This indicates that the abrasive wear rate is largely independent of the number of cycles undertaken. In all these experiments the contact conditions were little changed by the wear process, the contact patch sizes (and hence wear track width) were similar throughout the test.

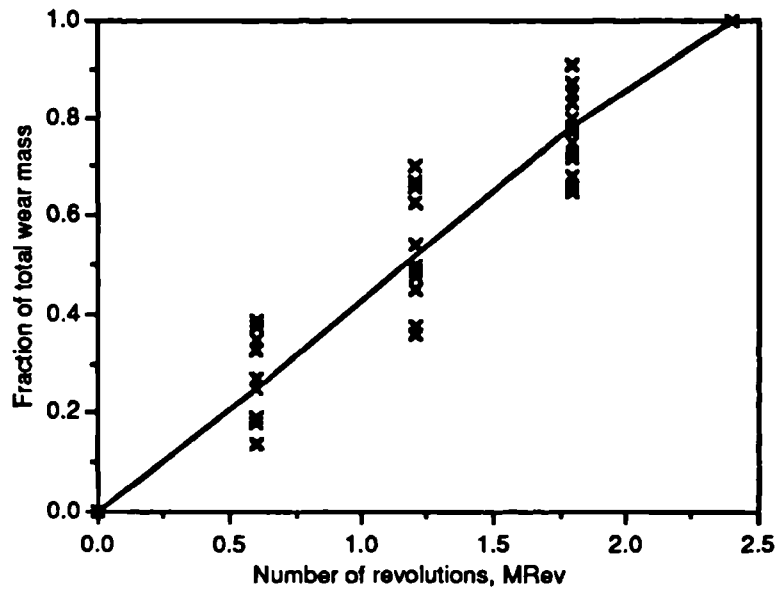


Figure 7.30. Fraction of bearing mass loss with the number of cycles. Data from all the tests is plotted as crosses; the average of all the data is plotted as a line.

The wear model, presented in the next chapter, assumes that the contact patch size does not vary with time, if this is the case then the wear rate is independent of number of cycles. The generated debris, from these tests, is too small to cause its own abrasion so this will not result in an acceleration of wear. Figure 7.30 lends credence to these assumptions.

The data from the tests using Arizona dust are also included on Figure 7.30, and show the same linear relationship between wear and test cycles. Arizona dust particles fracture in the inlet region to the contact, thus it is possible that the supply of debris (which was not topped up at any stage) may move to a smaller particle size³⁰. However, it is suggested that wear is controlled by these final fragments (and largely independent of original particle size) and thus the fracture process would not affect wear rates; large particles would fracture on entry down to small particles in any case.

7.3.2.5. Mass Loss and Abrasive Concentration

The results of three tests at different concentration of 6 μ m silicon carbide are shown in Figures 7.31 and 7.32. Each point on the graph is an average of two data points (one for each of the test bearings in the apparatus).

The mass loss appears to vary approximately linearly with the debris concentration. This is in agreement with the theories proposed. Each particle contributes individually to the overall wear; so increasing the number of particles proportionately increases the amount of material loss.

³⁰ A simple calculation suggested that some 25% of the debris particles would be overrolled during the eight hour test.

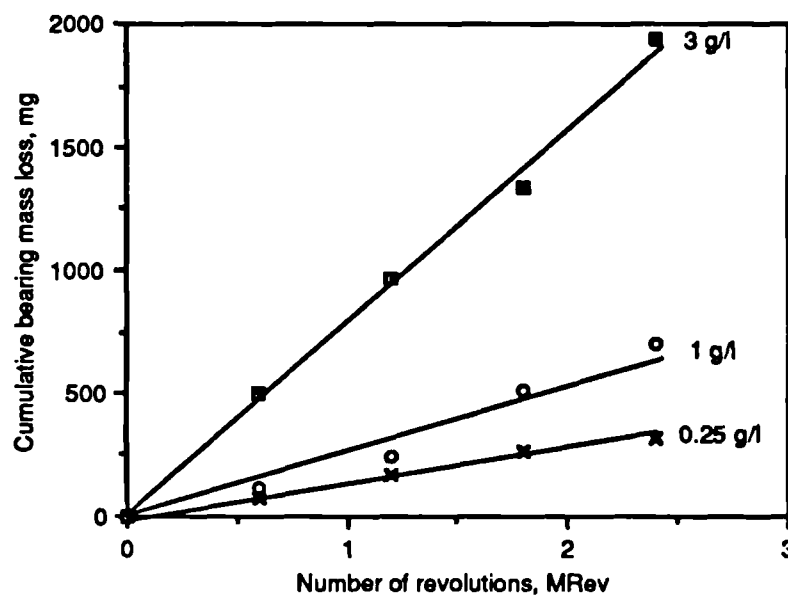


Figure 7.31. The cumulative bearing mass loss for three debris concentrations. Abrasive powder $6\mu\text{m}$ silicon carbide, bearing load 4500N.

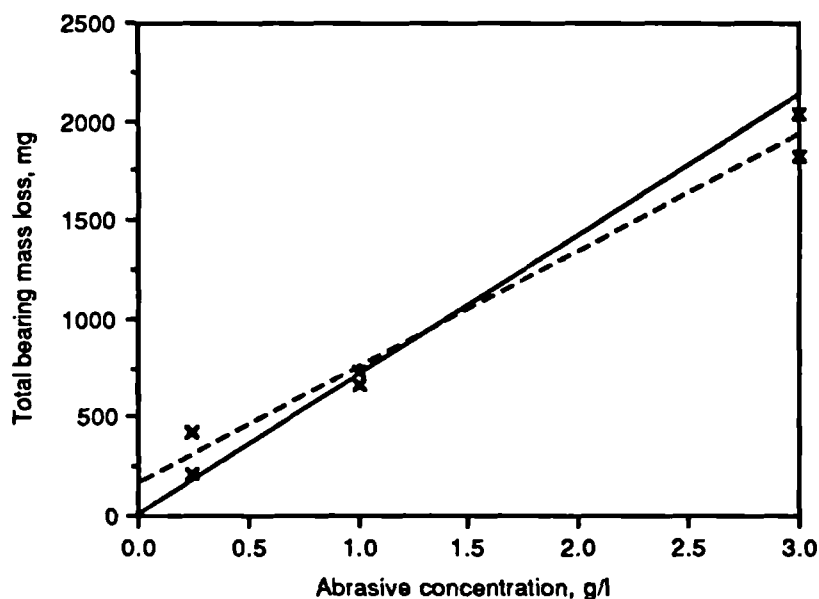


Figure 7.32. Total bearing mass loss plotted against the abrasive concentration in the tests. Abrasive powder $6\mu\text{m}$ silicon carbide, bearing load 4500N, test duration 8 hours.

It is interesting to note that the mean line drawn through the data (shown as dotted) does not pass through the origin. It is possible that some proportion of the debris is not being passed through the test bearings; but is instead lost in the oil circulation system. Hence, the bearings may show a zero mass loss at a finite debris concentration.

7.3.2.6. Bearing Wear and Abrasive Particle Size

In one test the debris used was 58-63 μm Arizona test dust (obtained by sieving). The results from this analysis are compared with those under similar conditions but using 6 μm silicon carbide as an abrasive. The graph included as Figure 7.33 shows the cumulative wear mass for the two different materials at the same concentration (again, the points are an average of data from two test bearings).

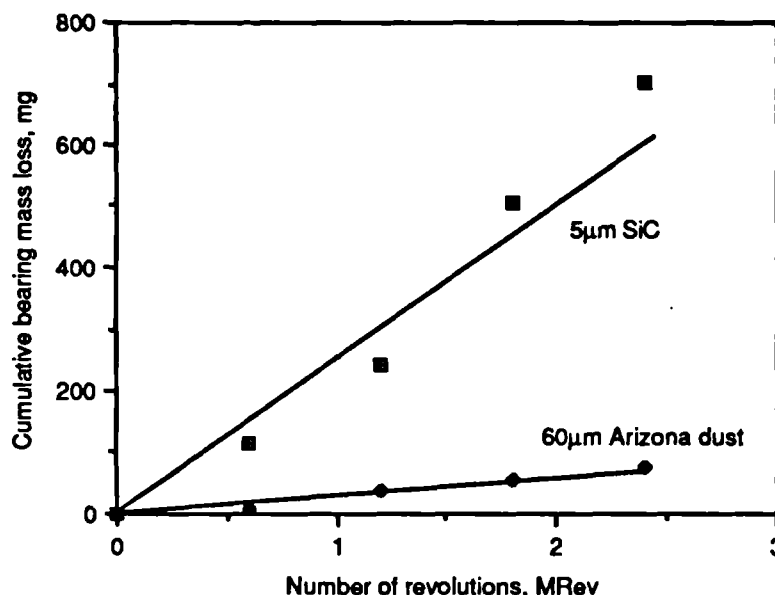


Figure 7.33. Cumulative wear mass with number of bearing cycles for two abrasive materials. Abrasive concentration 1 g/l, and bearing load 4500N.

Some degree of scatter is visible in the results; however about seven times the mass loss was found using the silicon carbide powder. At first it seems strange that the larger abrasive particle size should cause less wear. However, the Arizona dust is a less tough material and fractures readily in the inlet region, the final fragments are small, less than 1 μm in size. The silicon carbide particles do not appear to breakdown.

These 1 μm particles have a smaller cutting area; but also, it is likely that they tumble through the contact; whereas the 6 μm particles plough. The former leads to greatly reduced wear.

The model, presented in the next chapter, describes how a particle is imbedded between two surfaces separated by a film. A larger particle therefore causes more wear; however in a fixed concentration by mass, there are many fewer larger particles. The relation between particle size and mass loss is a compromise between the cutting area of the particle, the number of particles present in the lubricant, and the threshold for a tumbling wear mechanism (these relations are graphically demonstrated in §8.5.4).

7.3.2.7. Bearing Wear and Load

Tests were performed at a reduced load; mass losses were measured in the same manner. The scatter in the results (described above) made it difficult to form any general opinions from the limited data (two test bearings at each load). Figures 7.34 shows the cumulative wear mass for

bearings at high and low load using 1g/l of 6 μ m silicon carbide (again the data for the two bearings has been averaged the extreme results differ by about 30% from the mean).

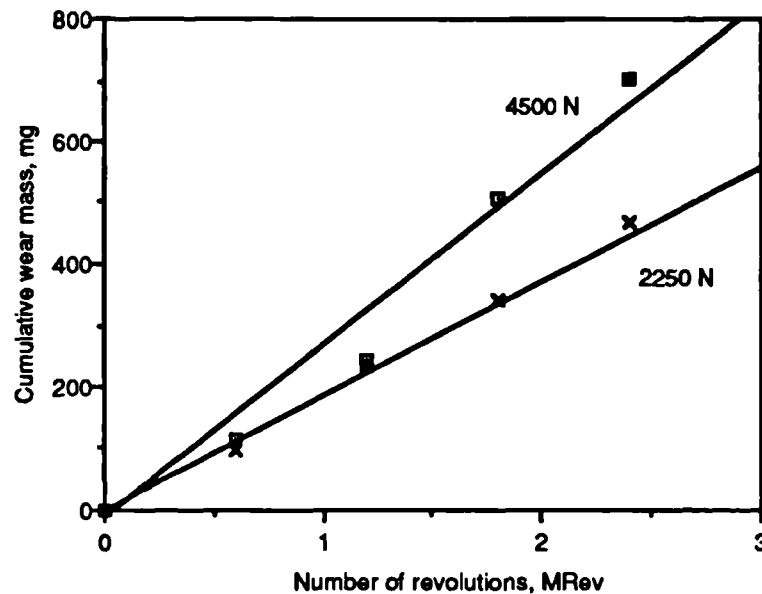


Figure 7.34. Cumulative mass loss of whole bearing at two loads, 2250 and 4500N. Abrasive concentration 1g/l of 6 μ m silicon carbide abrasive.

The results indicates that mass loss reduces slightly with reduced load; the effect is somewhat masked by the scatter in the results.

The effect of load on abrasive wear is secondary. Higher loads do not cause deeper particle imbedment (as in two-body abrasive wear); the particles become fully imbedded at even the smallest loads. Rather, an increase in load causes an increase in contact size which allows more particles to become entrapped. The film thickness variation with load is small and has little effect.

It is instructive to consider a single element of the bearing, the inner raceway, and look at the change in wear under different load. Averaging all the data for all the measurement intervals gives a ratio of inner raceway mass loss at 4500N to 2250N of 1.2 (sample size 8). The wear model (described in the next chapter) predicts a the mass loss to be proportional to the load to the power of 4/3 (see §8.5.3). Therefore, doubling the load would be expected to increase the wear mass by a factor of 2.5. Some discrepancy is evident and is attributed to the lack of statistical data; more tests are required to achieve a sound analysis.

7.3.2.8. The Effects of the Cage on Ball Wear

The plastic cages in these bearings are not a close tolerance component; so the ball cage contact conditions may be slightly different from one ball to the next. In one test the balls were kept in the same cage cup position throughout. The test was performed with the plastic cage specimens (re-assembly of steel cages was not possible). The mass lost from each ball was recorded at each measuring interval. Figure 7.35 shows a plot of the results.

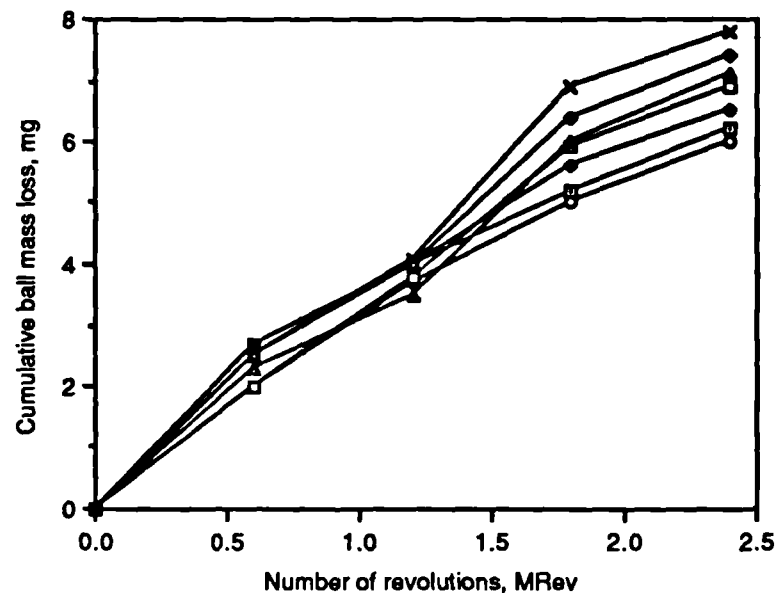


Figure 7.35. Cumulative mass loss from each ball against number of revolutions (each line represents a single ball). Concentration 1g/l Arizona dust, load 4500N, plastic cage.

The balls do show some different wear behaviour, the mass lost varies by about 20%. The graph included as Figure 7.36 displays the final mass loss data in a different format. The numbering of the balls follows the sequence that they were installed in the cage (i.e. 1 is next to 2 and so on).

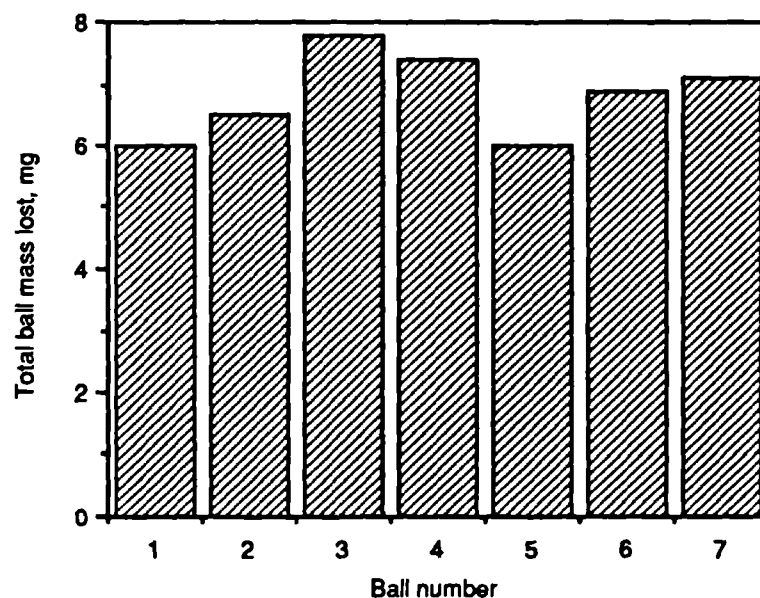


Figure 7.36. Total mass loss for each ball. Concentration 1g/l of Arizona dust, load 4500N, plastic cage.

The variation in ball wear is slight, the test does not conclusively prove that the cage has an effect. It is more likely, given the minor role played by the cage, that it is some micro-structural

or hardness difference between the balls which results in this effect. Particularly when one considers how small hardness differences can dramatically affect wear.

It should be noted that these wear masses are small compared with the total mass of the ball (some 0.2%). It is therefore unlikely that these different ball wear rates could lead to any significant out of balance mass in the assembled bearing.

7.3.2.9. Bearing Diametral Clearance and Abrasive Wear.

A worn bearing might fail, for example, by excessive vibration. The loss of material leads to an increase in clearance in the bearing and vibration occurs. Therefore, a more practically useful measure of abrasive wear is the diametral clearance in the bearing.

Figure 7.37 shows the variation in diametral clearance throughout a test, for one of the bearing specimens.

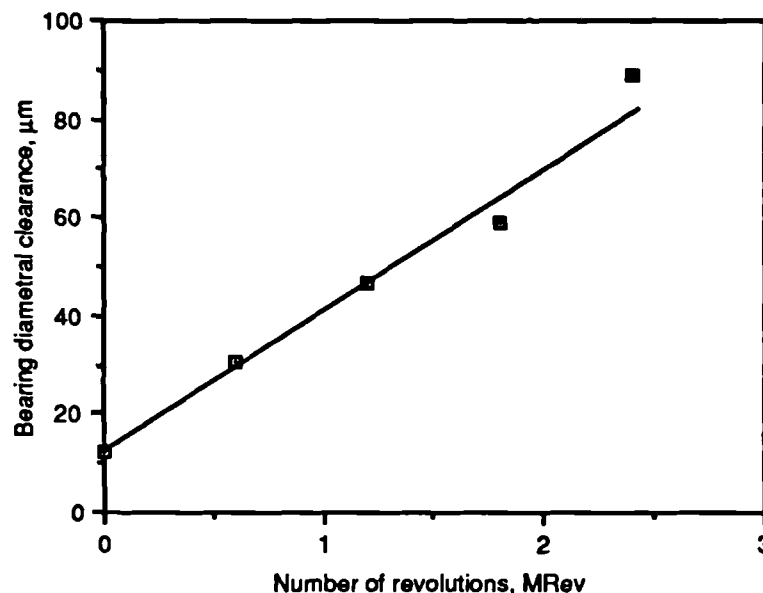


Figure 7.37. Bearing diametral clearance variation against number of bearing revolutions. Abrasive concentration 1g/l silicon carbide, load 4500N, plastic cage.

The bearings in their unworn conditions have a small clearance (ranging from $4\mu\text{m}$ to $15\mu\text{m}$). As with the wear mass, the clearance increases approximately linearly with number of revolutions.

Figure 7.38 shows that the clearance also increases approximately linearly with the concentration of debris in the lubricant. This would be expected if the mass loss remains linear with time and no variation in contact width results. The dotted line shows a straight line fit to the data. Again, the line does not pass through the origin, suggesting that some of the debris may be lost in the system.

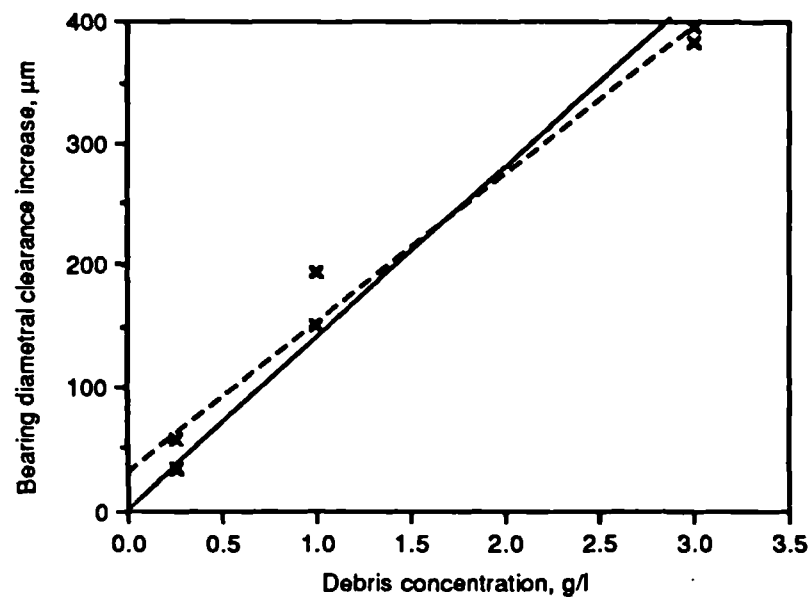


Figure 7.38. Increase in bearing diametral clearance against concentration of debris. Silicon carbide abrasive, load 45000N, plastic cage.

§8.4.2 describes a method for calculating the diametral clearance from the wear masses of the individual components. Table 7.8 compares this calculated clearance with the actual measured clearances.

Specimen	Number of Revolutions, MRev	Calculated Clearance, μm	Measured Clearance, μm
A	0.6	19	18
A	1.2	48	35
A	1.8	59	40
A	2.4	73	68
B	0.6	52	43
B	1.2	76	58
B	1.8	102	78
B	2.4	132	110

Table 7.8. Bearing diametral clearance predicted from the mass loss compared with the measured value. Abrasive concentration 1g/l silicon carbide, load 4500N, plastic cage.

This results show reasonable agreement (within 20%) with some slight over-prediction. The reasons for this are discussed in §8.4.2.

It is interesting to note how the wear of each element contributes to the overall increase in diametral clearance. The pie chart included as Figure 7.39 represents the proportion of the increase in clearance attributed to each component³¹ (as calculated using equations 8.65, 8.66, and 8.67 from §8.4.2).

³¹ This data is from a test using silicon carbide abrasive. in §7.3.2.1 it was explained that the ball wear, and therefore clearance attributable, was less for arizona dust abraded specimens.

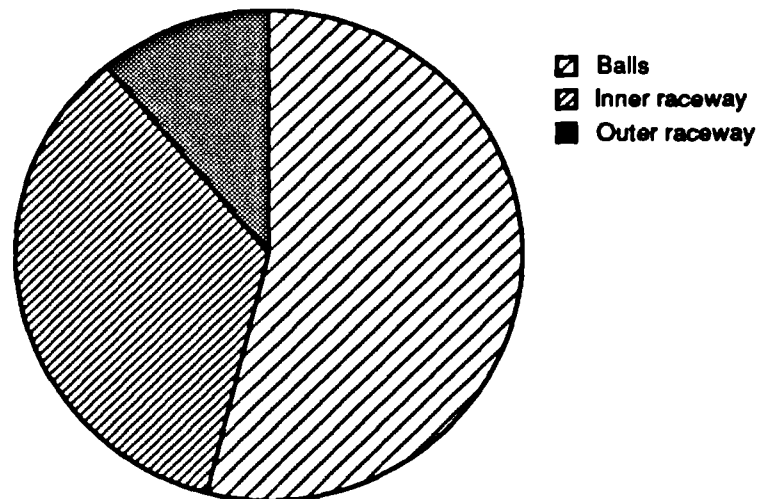


Figure 7.39. Pie chart showing the diametral clearance increase attributable to each of the elements of a ball bearing abraded with silicon carbide.

It is instructive to compare this chart with the equivalent for the wear mass attributable to each bearing element (Figure 7.21). Although the balls dominate in mass loss, their effect on bearing clearance increase, and therefore bearing life, is less prevalent. This is because the balls may wear over their whole surface³² and the clearance is not increased as much as if material was removed from a single circumferential band.

If one were to design a 'smart' bearing of maximum wear life, defined in terms of clearance, it would therefore be desirable to have more wear on the balls.

7.4. Comparison with Other Data

Little published material is available on wear in rolling bearings (or indeed any other closed three body wear process). In many cases it is difficult to compare wear rates; different specimens, debris materials, and contaminating procedures were used.

However the following data has been analysed and is presented alongside the most closely comparable results of this study.

7.4.1. Loewenthal & Moyer (1979)

Lowenthal and Moyer performed experiments to test the effect of filtration on bearing lives under conditions of contaminated lubrication. Five series of tests were performed with increasingly coarse filtration; only the fifth series is of interest here. The other filtration levels were too effective and kept wear down to a negligible amount.

In the fifth series wear rates of approximately 3mg/MRev were measured. Particle count readings suggests about 1.6g/l of debris were being passed through the test bearings. The contaminants used were a mixture of 90% graphite (0-20µm), 9% Arizona dust (0-100µm),

³² Although in some cases tracking of the balls was noted. In this case the wear is not shared equally over the whole ball surface and the increase in clearance resulting from the ball wear is greater.

and 1% stainless steel ($<44\mu\text{m}$). The test bearings were about three times the size of those used in the present study.

In the present study the closest comparable data are wear rates of between 9 and 20mg/MRev for 1g/l Arizona dust contamination. At first site the wear rates of Loewenthal and Moyer seem low. However it is likely that the graphite debris materials are causing little mass loss. It is suggested that the majority of the wear is caused by the Arizona dust particles. Bearing this in mind, the data are not inconsistent.

7.4.2. Skorynin & Minchenya (1984)

This largely theoretical paper provides one piece of experimental evidence to compare with this study. The breakdown in mass loss between the components of a ball bearing abraded with corundum are recorded. The pie chart included as Figure 7.40 displays this data.

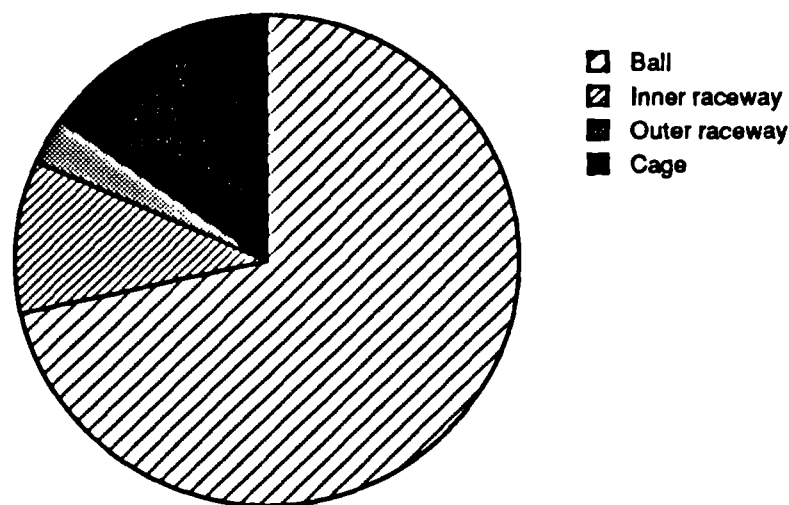


Figure 7.40. Pie chart of the breakdown in mass loss between the individual components of a ball bearing; data from Skoryin & Minchenya (1984).

This breakdown corresponds closely with the findings in this study (see Figure 7.21). The bulk of the mass loss has been from the balls. The originators of the data suggest this is caused by the abrasive action of debris trapped in the cage. It is this authors opinion that this is not the major reason. It is possible that some form of preferential damage is occurring to the balls in a ball/raceway contact. The particles were $6\mu\text{m}$ in size and so the mechanism was probably one of ploughing; rather than tumbling which would have led to a more even distribution of wear (see Figure 7.22 and §7.3.2.1).

7.4.3. Fitzsimmons & Clevenger (1977)

The study of Fitzsimmons and Clevenger provides some of the most useful comparative data. The test bearings were tapered rollers, and therefore not directly relevant; however contact areas and film thicknesses were of the same order.

The first point to note is the scatter in the data; typically results vary by as much as 50%. In their study several bearings were tested for each condition; the data is always presented statistically with a correlation coefficient. The same sort of scatter was observed in this study.

Mass loss was found to be linear with contaminant concentration; this was also the case in these experiments (see Figure 7.30). Using silica as a contaminant with a concentration of 1g/l the mass loss was about 135 to 160mg in 10 hours. The data for similar conditions in this study, using Arizona dust, was a mass loss of between 45 to 100mg in 8 hours; a good agreement is seen.

Fitzsimmons and Clevenger found that with long duration tests of 40 hours, using silica as an abrasive, the wear rate tailed off. 90% of the wear occurred in the first 10 hours of the test. It was suggested that the silica particles were becoming crushed down. Adding further contaminant caused a renewed cycle of wear. In the present study, tests were not continued past 8 hours, and so no evidence of this levelling off was noted. Even the crushed down particles would cause some wear since they are likely to be larger than the film thickness ($0.2\mu\text{m}$); indeed the studies presented in this thesis suggest that wear is largely controlled by these fragments. It is possible that the tailing off may be due to the particles dropping out of suspension.

Also presented, were some interesting experiments on the effect of lubricant viscosity on abrasive wear. Figure 7.41 is a replotting of their data, in terms of lubricant film thickness against bearing mass loss. As the film thickness increases, more of the particle can be accommodated without surface damage, and the wear is reduced.

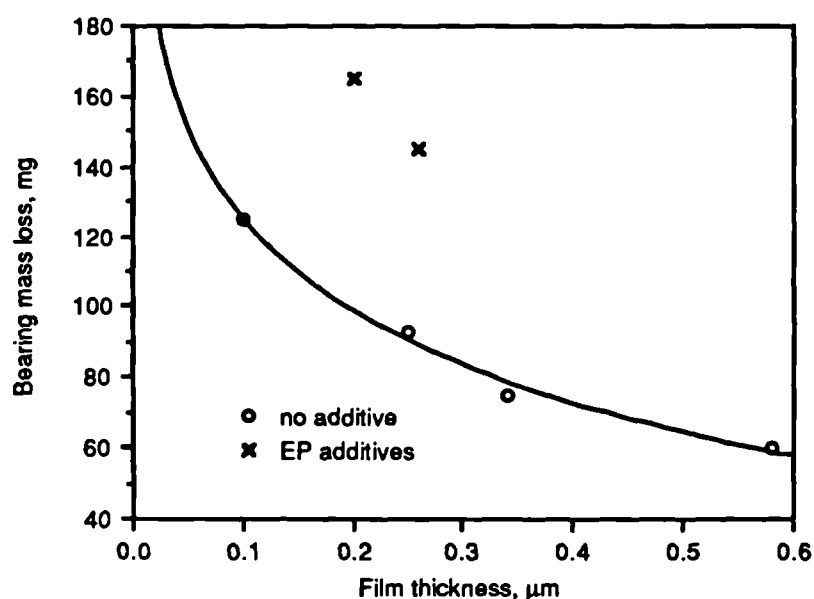


Figure 7.41. Bearing mass loss against calculated lubricant film thickness, 1g/l silica abrasive; data from Fitzsimmons and Clevenger (1977).

Where EP additives were present in the oil the mass loss was higher than expected. The additive works by chemically attacking the metal surface. The abrasion process provides fresh surface after each action; this coupled with the additive action leads to increased wear.

7.4.4. Desert Helicopter Rotor Bearings

Bearings from the tail rotor of a helicopter used under desert conditions have been studied for possible abrasive wear damage (Dwyer-Joyce 1992). Several specimens of various service life were available for analysis. The surface appearance was somewhat similar to those bearings

abraded by Arizona dust, although less severe. Initial masses were not known; however diametral clearances could be measured. Figure 7.42 shows a plot of the clearance measurements for five bearings against their service life (all were removed before their full service life was expired).

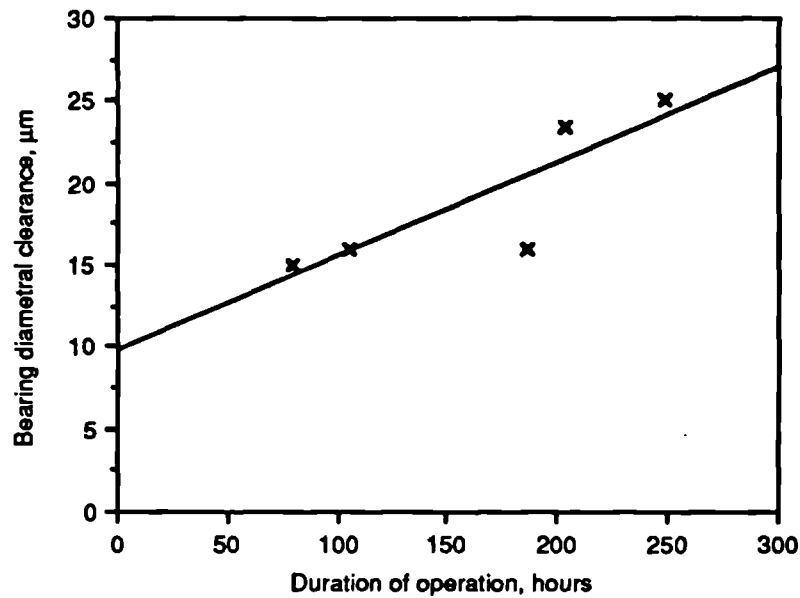


Figure 7.42. Diametral clearance plotted against service hours for helicopter tail rotor bearings operated under desert conditions.

The data shows a trend towards increased clearance. Given that the bearings were smaller than those used in these tests, the clearance increases are not incomparable.

7.4.5. The Wear Coefficient k

In the Archard (1953) wear equation (see §6.3 equation 6.1), and the modification by Rabinowitz (1965) the wear, V is proportional to the load, W times the sliding distance, l , divided by the hardness, H . The constant of proportionality, k is the wear coefficient;

$$V = \frac{kWl}{3H} \quad (7.8)$$

The usual approach in wear prediction is to perform some experiments on a given system and determine the wear coefficient from the results and use this value as a constant for that system.

In this closed three body wear mechanism the debris particles become fully imbedded into the rolling elements. The load is therefore defined by the hardness of the elements and given by;

$$W = \frac{\pi\delta^2}{4} H \quad (7.9)$$

where δ is the diameter of an assumed spherical debris particle. Putting this into Archard's equation gives a wear equation of the form;

$$\frac{dV}{dl} = k \frac{\pi \delta^2}{12} \quad (7.10)$$

The wear volume, V caused by a single abrasive grain is given by the total volume lost from the bearing divided by the number of overrolled grains. The sliding distance, l is an average value determined from the microslip of a ball on raceway contact, d_{av} . And δ is an average particle dimension. These values are readily calculated from the test conditions and results. The calculation was repeated for the ball, inner raceway, and outer raceway wear sites. So, rewriting equation (7.10), if M is the total wear mass for one of the elements where N particles were entrained, then;

$$M = kN d_{av} \frac{\pi \delta^2}{12} \quad (7.11)$$

The wear coefficients obtained from the bearing test data are in the range 5×10^{-3} to 15×10^{-3} . The silicon carbide and Arizona dust data showed little difference. Table 7.9 compares this data with that of other investigators.

Investigator	Wear Classification	Abrasive	k ($\times 10^{-3}$)
Spurr et al (1957)	2-body	files	180
Spurr et al (1957)	2 1/2-body	emery	150
Avient et al (1960)	2 1/2-body	emery	120
Lopa (1956)	2-body	grinding wheel	80
Khrushov (1957)	2 1/2-body	corundum paper	24
This study	closed 3-body	SiC & Arizona dust	5-15
Toporov (1960)	open 3-body	Al ₂ O ₃	6
Rabinowitz et al (1961)	open 3-body	Al ₂ O ₃	4.5

Table 7.9. Abrasive wear constants determined using Archard's wear equation from the work of various investigators.

The two body wear mechanism gives highest wear coefficients; the abrasive is generally rigidly fixed and all its energy is expended in a wear process. In the two and a half body case the abrasive is flexible held on a backing paper, wear is reduced. In the open three body case abrasives are free to move or roll and wear is further reduced. In this study the process is closed three body abrasion; the particles are trapped between two surfaces, and the wear coefficient falls somewhere in between.

7.5. Conclusions

A ball on disk geometry has been used to investigate some of the fundamental aspects of abrasive wear. Examination of the surface damage has been used to predict the motion of the abrasive particle when trapped between the two elements. A bearing test rig has been used to generate some information on the wear of ball bearings by lubricant debris particles. Both sets of tests were performed using small size ceramic particles as abrasives. This removal of the complexities involving particle deformation, allowed for a more fundamental study on bearing abrasive wear.

The following conclusions have been drawn from the ball on disk experiments.

- (a) When using standard bearing components the slightly harder balls were grooved whilst the disks were only pitted. All of the relative sliding was accommodated by the particles imbedding in the softer disk surface and scratching the ball. The reasons for this behaviour are not fully obvious.
- (b) The lengths of the grooves corresponded closely with the sliding distance of the abrasive particles.
- (c) Changing the hardness ratio of the elements altered the location of the grooves. Although a clear relationship was not evident, possibly due to the tempering process used to soften the specimens.
- (d) When the particle size to film thickness ratio was less than about 2 the particles tumbled through the contact, and pits were formed on both elements. This mechanism would lead to greatly reduced wear.
- (e) The scatter in the measurements of the geometry of the abrasive grooves was likely to be caused by the size range of abrasive particles.
- (f) The geometry of the abrasive groove was used to estimate the proportion of the area displaced by an abrasive particle which becomes removed as a wear chip. This factor, f can then be used to predict the wear volume caused by a single abrasive particle.

The following conclusions have been drawn from the bearing abrasive wear experiments.

- (a) The mass lost from the balls tended to dominate the wear of bearings tested with $6\mu\text{m}$ silicon carbide abrasive. The particles were thought to imbed in the softer raceways and preferentially scratch the ball surfaces, in a manner similar to that noted in the ball on disk tests.
- (b) When bearings were tested with Arizona dust the mechanism was thought to be one of particle tumbling. The wear was considerably lower and more evenly distributed between the components of the bearing. The ratio of the abrasive fragment size to oil film thickness was less than 2.
- (c) The plastic cages used in the tests were thought to contribute little to the overall wear of the bearing.
- (d) Wear was found to be directly proportional to the debris concentration, and to the number of cycles. The data suffers from a high degree of scatter; a statistical approach to this sort of wear testing is appropriate.
- (e) The variation of wear on the raceway surfaces was found to correspond closely with the pattern of microslip within the contact.
- (f) The bearing abrasive wear data of other experimenters is largely consistent with this work.

These bearing tests have shown some of the wear mechanisms highlighted in the ball on disk tests. The model described in the next chapter uses this information to predict bearing abrasive mass loss.

CHAPTER 8

ABRASIVE WEAR MODELLING

8.1. Introduction

The optical ehl experiments have shown how particles behave in and around a contact. Estimates have been made for predicting the number and size of debris particles that may enter the contact. The simple tests using a ball on disk contact have shown how these particles can cause material removal. Depending on the conditions, particles were found to tumble or plough through the contact. The hardness of the rolling elements was found to be of singular importance in controlling the extent of surface damage. Special attention has been paid to determining the amount of material removed by a single abrasive particle as a function of its size. The bearing tests have shown that these mechanisms are important in the consideration of rolling bearing wear.

Several models exist for two-body abrasive wear, and are applicable for machining or abrasive paper type problems. Some of the concepts from these models can be useful for predictions of three body wear.

At this stage, we are in a position, to bring these elements together and form a model to predict rolling bearing wear under certain conditions of lubricant contamination.

8.2. The Mechanism of Closed Three-Body Abrasive Wear

In a two-body abrasive wear process an abrading element penetrates the surface of another body and undergoes some motion relative to that body. This motion results in a ploughing action and the formation of a wear chip or built up shoulders which are removed later. If the particle is trapped between two surfaces (i.e. closed three-body abrasive wear) both surfaces may be ploughed; or the particle may imbed in one and cut the other.

In the Rabinowitz (1961) model of abrasive wear, the particle is modelled as a cone which imbeds into one surface and ploughs out a groove. The load and the geometry of the grit control the cutting area and hence the volume of material removed. Rabinowitz derives an equation;

$$\frac{dV}{dl} = \frac{W \cot \theta}{\pi H} \quad (8.1)$$

where V is the wear volume which is defined per unit sliding distance, l . W is the load and H is the hardness of the abraded material. The average angle of the conical abrasive particles is θ .

In this closed three-body wear mechanism the imbedment of the abrasive particles (and hence wear volume) is not controlled by the load on the particles. The load on the rolling elements is high and any particle entrained between them becomes fully imbedded, with a film thickness separator. Figure 8.1 compares the two processes. The mechanism is thus essentially displacement, rather than load, controlled.

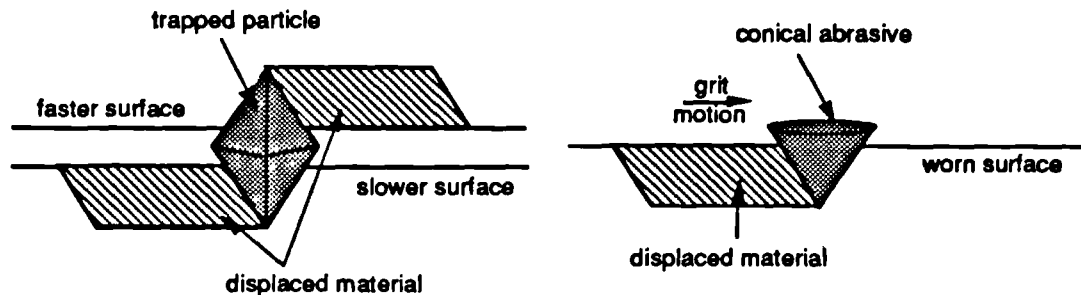


Figure 8.1. Abrasive wear mechanisms for (a) closed three-body and (b) two-body abrasive wear. In the former case the depth of indentation is displacement controlled, whilst in the latter it is load controlled.

The closed three body abrasive wear process is, thus, a special case of abrasive wear. If the size of the lubricant debris particle can be quantified then we know it will be fully imbedded. We can determine sliding distances and film thicknesses from the kinematics of the rolling bearing. This can be coupled with an understanding of some of the mechanisms of abrasive wear to create a model of the process.

The model described here is limited to ball raceway interactions. The role of the cage is discussed in §8.6.3. The bearing tests described in Chapter 7 showed that the cage had a minor effect on the overall wear of the bearing.

8.3. The Important Parameters

From considerations of the three body abrasive wear process, it is apparent that to predict wear mass several basic parameters are required;

- (i) The Sliding Distance. The sliding within a ball on raceway contact is caused by the Heathcote, or microslip, within the nominally rolling contact. This is readily defined from the kinematics of the ball bearing.
- (ii) The Particle Cutting Area. In this closed abrasive wear situation the particle is assumed to be rigid and imbeds fully into the two opposing faces. The cross sectional area of the imbedded particle will displace material from the surface in relative motion. This cutting area can be defined from the lubricant film thickness and an assumed particle geometry.
- (iii) The Number of Particles Overrolled. The kinematics of the bearing defines the volume of lubricant overrolled in a given time period. It is assumed that the number of particles that enter a contact is in proportion to their concentration in the bulk. (This particle entry model is described and further justified in §2.4.4.2)

- (iv) The Proportion of the Sliding Distance Accommodated on the Wearing Surface. The ball on disk tests showed that depending on the hardness of the rolling elements one surface can hold the particle which then scratched the other. The mass loss is thus not equally proportioned between the two; a factor, g is defined to model this behaviour.
- (v) The Proportion of the Groove Volume Removed as a Wear Chip. At the heart of the wear process is the requirement for understanding how much material each particle removes as wear. Some of the material displaced by the particle will be accommodated elastically, some by the formation of shoulders, and some will be removed from the surface as a wear chip. The definition of the proportion removed as a wear chip has been approached empirically. Data from the bearing tests and the ball on disk tests have been used to quantify this grooving factor, f .

At its simplest the wear volume is a multiplication of these four factors. However, other features complicate the process somewhat. Experiments have shown that small particles can tumble, rather than plough, through the contact; thus a fifth factor is added;

- (vi) Threshold for Particle Tumbling. The equilibrium of a particle trapped between two surfaces is used to predict when a transition from ploughing to tumbling will occur.

In the next six sections each of these factors is discussed and the mathematics explained.

8.3.1. Sliding Distance and Bearing Microslip

The sliding distance in a ball on raceway contact is determined from the microslip in the nominally rolling contact. The first solution to the theory of slip and rolling resistance to a ball rolling in a conforming groove was presented by Heathcote (1921).

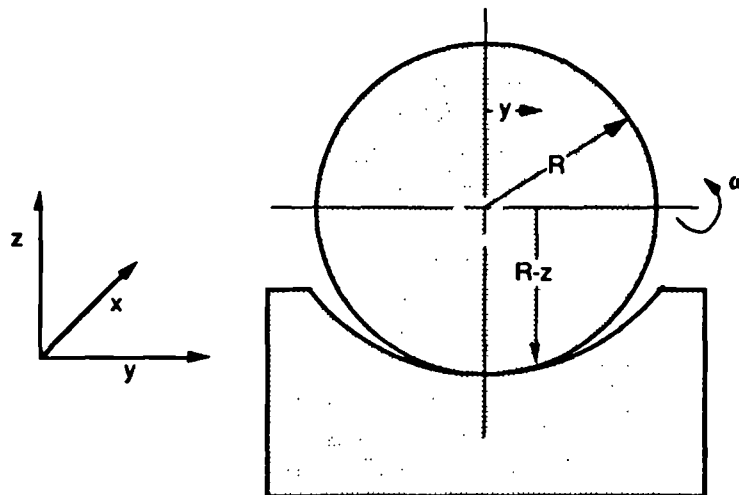


Figure 8.2. Diagram of ball on raceway contact showing co-ordinate system and the rolling radii.

Referring to Figure 8.2, the peripheral velocity of the ball, V_1 is given by;

$$V_1 = \omega(R - z) \quad (8.2)$$

where ω is the angular velocity of the ball of radius, R . The co-ordinate variables y and z are defined on Figure 8.2. If z is small then to a reasonable approximation;

$$V_1 = \omega \left(R - \frac{y^2}{2R} \right) \quad (8.3)$$

The slip or *creep ratio* , ξ is defined by;

$$\xi(y) = \frac{V_1 - V_2}{\omega R} \quad (8.4)$$

substituting for V_1 gives;

$$\xi(y) = \left(1 - \frac{V_2}{\omega R} \right) - \frac{y^2}{2R^2} \quad (8.5)$$

at the axis of the ball, when $y=0$, the creep ratio, ξ_0 is given by;

$$\xi_0 = \left(1 - \frac{V_2}{\omega R} \right) \quad (8.6)$$

substituting into equation (8.5) above gives;

$$\xi(y) = \xi_0 - \frac{y^2}{2R^2} \quad (8.7)$$

For a condition of free rolling there is no net tangential force exerted on the ball. The slip regions must be split into three, a central zone and two outer zones with the slip in opposite directions. There are therefore regions of no-slip; if the regions are defined at $y=\gamma b$, then;

$$\xi_\gamma = \xi_0 - \frac{\gamma^2 b^2}{2R^2} = 0 \quad (8.8)$$

and so substituting for ξ_0 in equation (8.7) gives;

$$\xi(y) = \frac{\gamma^2 b^2 - y^2}{2R^2} \quad (8.9)$$

The constant, γ can be found by using the no net tangential force condition;

$$T = \iint \mu p \delta x \delta y = 0 \quad (8.10)$$

where $p(x,y)$ is the pressure distribution over the contact area, and μ is the friction coefficient which is assumed to be constant. the pressure distribution, p over an elliptical contact is given by Hertzian analysis;

$$p = p_0 \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}} \quad (8.11)$$

which is valid for $x \leq a$ and $y \leq b$, and where p_0 is the maximum pressure occurring at $x=y=0$; and a and b are the contact dimensions in the x and y directions respectively. So the condition becomes;

$$T = \iint \mu p_0 \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}} \delta x \delta y \quad (8.12)$$

$$T = \int \int_{y=0}^{y=b} \sqrt{1 - \frac{x^2}{a^2}} \mu p_0 \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}} \delta x \delta y \quad (8.13)$$

Integrating this gives;

$$T_{y_1-y_2} = \frac{\mu p_0 \pi a}{2} \left[y - \frac{y^3}{3b^2} \right]_{y_1}^{y_2} \quad (8.14)$$

The total traction is found by using the limits $-b$ to $+b$. To find the position of the pure rolling region (where the traction changes sign) the limits 0 to γb and γb to b are used and the two tractions equated, that is;

$$T_{0-\gamma b} = -T_{\gamma b-b} \quad (8.15)$$

Solving this gives;

$$\gamma = 0.35 \quad (8.16)$$

And so the creep distribution is given by;

$$\xi(y) = \frac{(0.35b)^2 - y^2}{2R^2} \quad (8.17)$$

This analysis takes no account of the ability of the surfaces to elastically deform to accommodate the relative slip.

A more rigorous analysis is presented by Johnson (1962) using Carter-Poritsky strip theory. The equations are stated here without derivation. Each strip of the contact is made up of regions of stick and slip. Figure 8.3 shows the regions of stick and slip at a strip of the contact of length a^* .

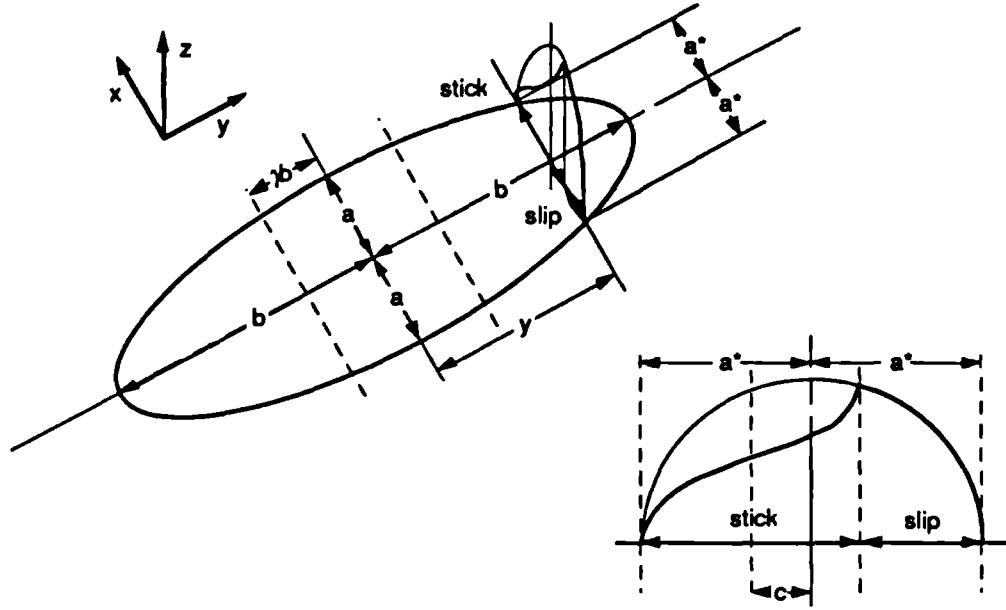


Figure 8.3. Regions of stick and slip in the contact; the cross section shows the conditions at a strip of length a^* .

The slip for this strip, located at y , and of length a^* , is given by;

$$\frac{c}{a^*} = \Gamma \left(\gamma^2 - \frac{y^2}{b^2} \right) \quad (8.18)$$

where Γ is the conformity factor defined by;

$$\Gamma = \frac{b^2 E^*}{4\mu p_0 R^2} \quad (8.19)$$

Again γ is determined by using the criteria for no resultant traction, T . The traction for the strip T^* is given by;

$$T^* = \frac{\pi}{2} \mu p_0^* a^* \frac{c}{a^*} \left(2 - \frac{c}{a^*} \right) \quad (8.20)$$

The total traction is zero so;

$$T = \int_{-b}^b T^* dy = 0 \quad (8.21)$$

Solving this integral between the limits 0, γb , and b gives the position of the no-slip bands, γ . There are two extreme cases;

- (a) High coefficient of friction (so Γ is small), which leads to a small region of slip at the trailing edge of the contact. The slip bands are located at $\gamma=0.5$.

- (b) Low coefficient of friction and a high degree of conformity. In this case almost all the points, off the pure rolling lines, are sliding. This was Heathcote's assumption, and in this instance $\gamma=0.35$.

Johnson (1985) provides a useful graph (Figure 8.16 in his text) showing the relation between conformity, Γ and rolling resistance, showing these regions of mostly stick and mostly slip.

For the bearings used in the bearing tests and estimating the friction coefficient as 0.05 gives a value of the conformity, $\Gamma=53.2$. This puts the regime into mostly slip and Heathcote's assumption holds³³. The Heathcote slip equation is more convenient³⁴ to use and it is this that is used in the abrasive wear model.

Using this value of γ equation (8.9) defines the slip at any position y along the width of the contact. The amount by which an entrained particle slides is given by this creep ratio times the rolling distance. Figure 8.4 schematically explains the relation between the creep ratio, rolling distance, and sliding distance.

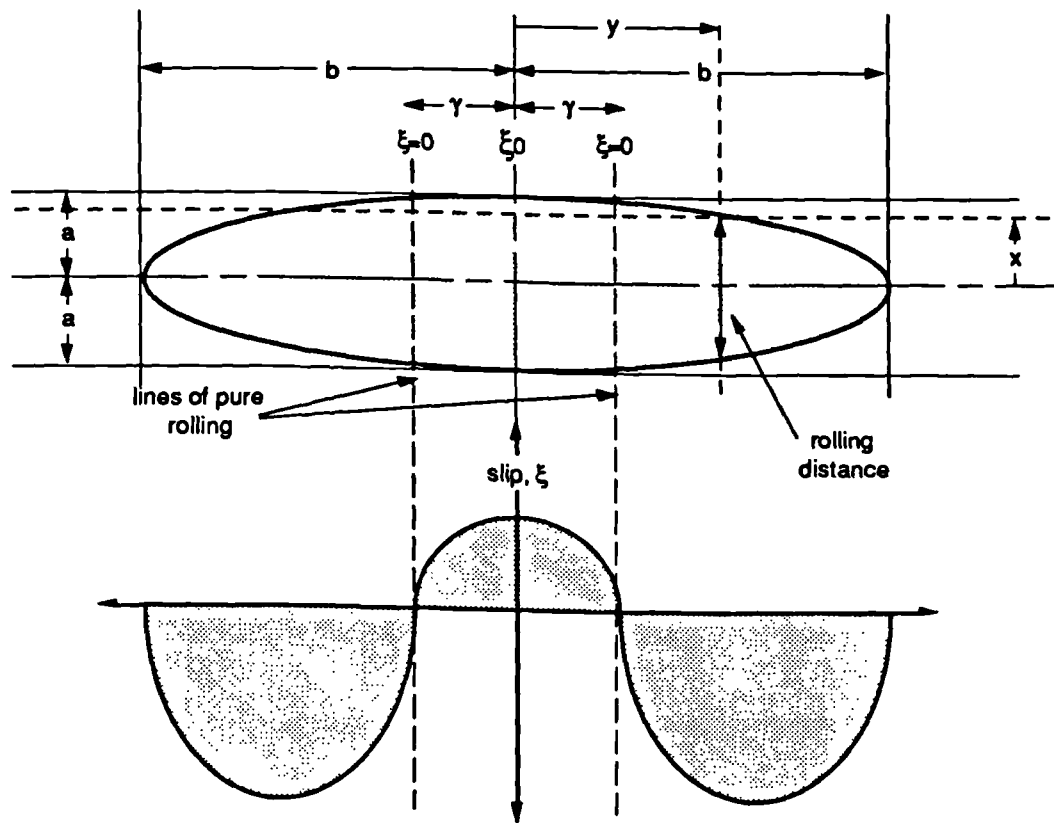


Figure 8.4. Diagram of slip regions in an elliptical contact; showing lines of pure rolling, at $y=\gamma b$, and regions of positive and negative slip.

The sliding distance, d of a particle, entering the contact at position y , is given by

³³ To check this statement, the full analysis was performed. This requires guessing a value of γ from equation (8.9) and performing the integration, numerically, from equation (8.21). The solution is iterated until the value of γ gives a zero total traction force. The analysis yielded $\gamma=0.362$ and indeed almost all of the contact was slipping.

³⁴ Since it is independent of friction and requires no iterative process.

$$d = \xi_y 2a \sqrt{1 - \frac{y^2}{b^2}} \quad (8.22)$$

substituting for ξ_y gives;

$$d = \left(\frac{\gamma^2 b^2 - y^2}{2R^2} \right) 2a \sqrt{1 - \frac{y^2}{b^2}} \quad (8.23)$$

The sliding distance is thus dependent on which location the particle enters the contact. Figure 8.5 is a plot of the rolling distance, creep ratio, and resulting sliding distance (defined as always being positive) plotted against the position, y in the contact.

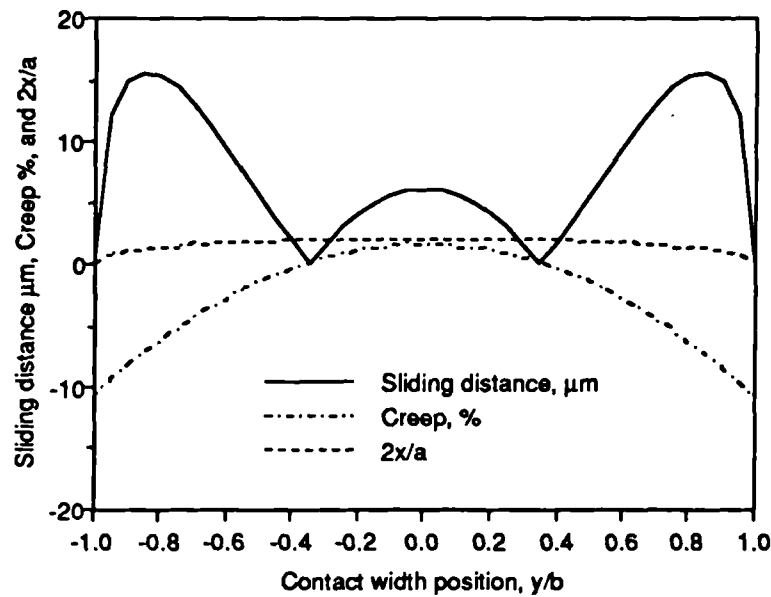


Figure 8.5. Rolling distance (normalised by the contact length a), creep ratio % (equation 8.17), and sliding distance in μm (equation 8.23)) against the position along the contact width, y/b . Data from a 6309 deep groove ball bearing.

In the abrasive wear process many millions of particles flow through the contact; statistically the average value of this sliding distance, d is useful. The wear can then be defined by an average sliding distance, d_{av} experienced by the entrapped particles.

This average value is determined by integrating for all possible entry paths, y across the contact patch $-b$ to $+b$, i.e.;

$$d_{av} = \frac{1}{2b} \int_{-b}^{+b} |d| dy \quad (8.24)$$

It is important to use the absolute value of the sliding distance in determining the average. Numerical integration³⁵, using $\gamma=0.35$ gives;

$$d_{av} = \frac{\pi ab^2}{20 R^2} \quad (8.25)$$

This gives an average value of the sliding distance for any single contact patch within the ball bearing. The wear generated by debris in a ball bearing does not occur at just one contact, but at all positions where a ball is in contact with a raceway. Since equation (8.25) is dependent only on the ball radius and contact patch size, it can be used for any contact within the bearing.

8.3.2. The Cutting Area of a Particle

At this stage an assumption is required about the geometry of an abrasive particle. Figure 8.6 shows a photomicrograph of 6 μ m silicon carbide particles.

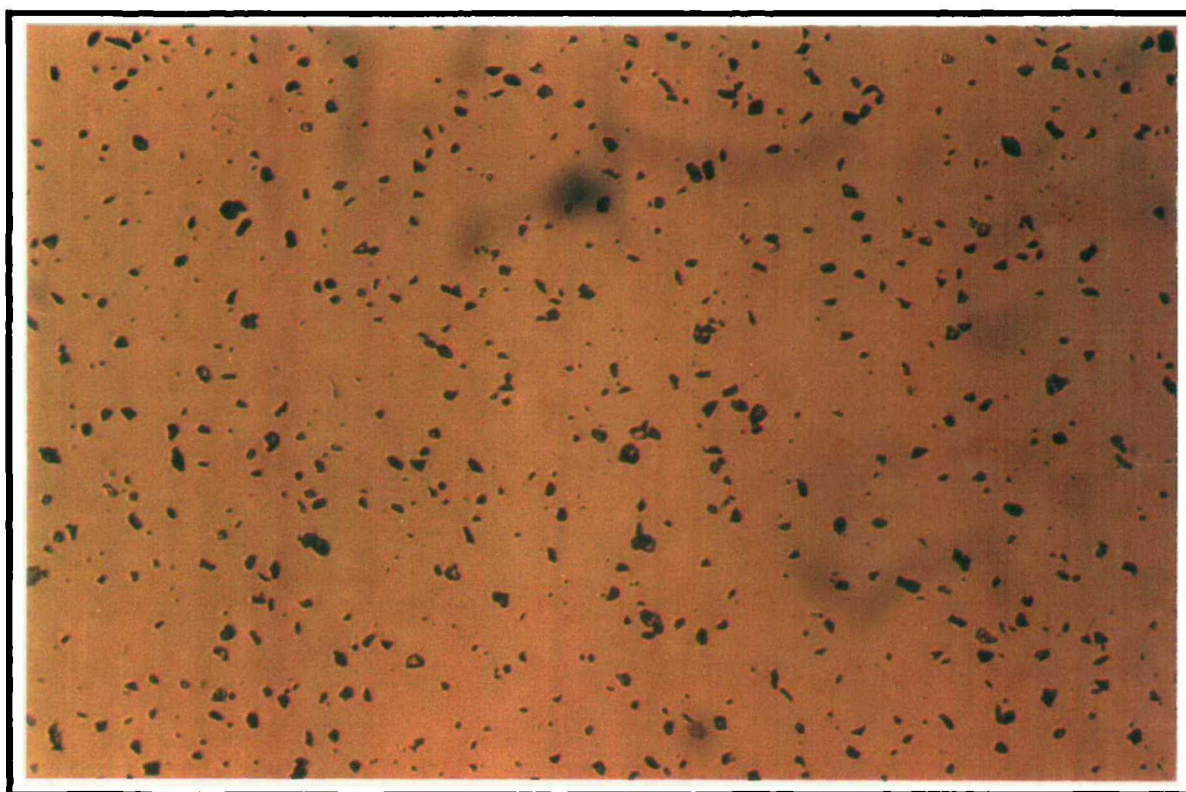


Figure 8.6 Photomicrograph of silicon carbide abrasive particles. Magnification x300.

Their appearance is 'blocky' with most included angles in the range 30°-120°, the ratio of the largest to smallest side length appears to be about 1.5. On the basis of this observation the model particle was chosen as a double pyramid. Figure 8.7 shows a diagram of the model particle. This symmetrical shape simplifies calculations; the model is generalised by use of the parameters δ_1 and δ_2 for side length, and ϕ for an included angle.

³⁵ A short program, using Simpson's rule, was written in FORTRAN 77 to perform this and other numerical integrations.

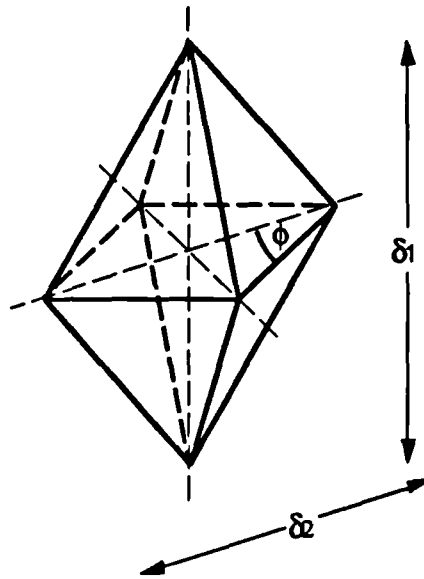


Figure 8.7. Diagram of the model abrasive particle.

This particle becomes imbedded between the two rolling elements, separated by a film of thickness, h . As relative sliding occurs between these two bodies the particle ploughs one, or both surfaces. It is the particle projected area transverse to the direction of motion, which is here termed the cutting area (or the displaced area), that controls the cross section of the ploughed groove. Figure 8.8 shows a diagram of the abrasive particle ploughing out a groove.

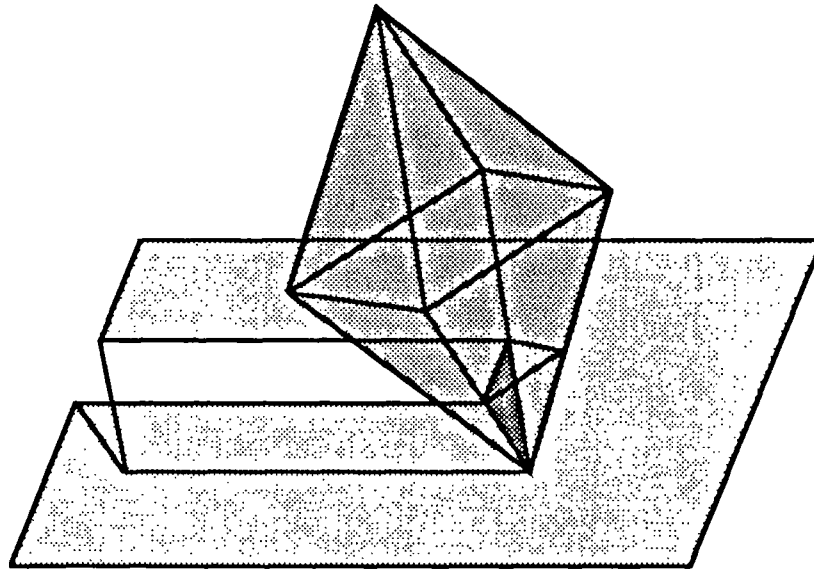


Figure 8.8. Diagram of the model abrasive particle ploughing a groove. The dark shaded cutting area controls the cross sectional area of the groove.

Since the abrasive grit is not fully symmetrical, the orientation at which the particle enters into, or equilibrates in, the contact will affect this cutting area. Figure 8.9 shows how both surfaces are ploughed when the particle is orientated at a generalised angle, α .

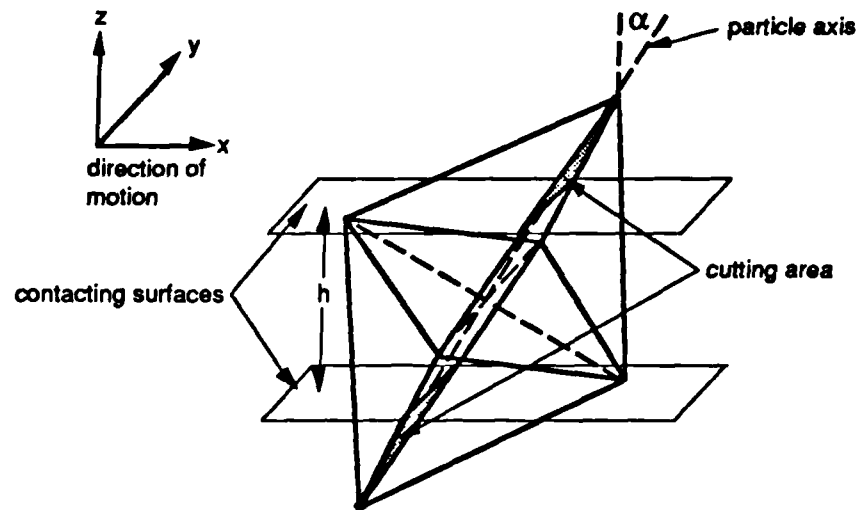


Figure 8.9. Diagram of the model abrasive particle in the contact. The two cutting areas are shown.

As the orientation of the particle changes the cutting areas change; for higher angles the second axis of the grit becomes dominant. Figure 8.10 shows the particle at various orientations, and how the major and minor axis intersect the two surfaces.

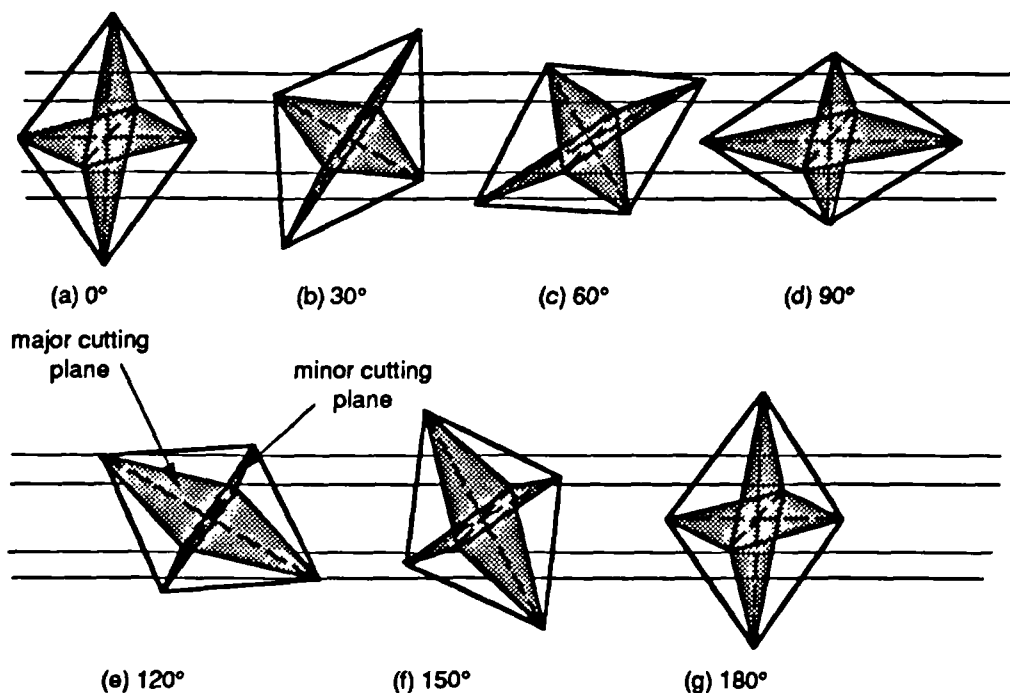


Figure 8.10. Diagram of the model abrasive particle at seven orientations in the contact. The major and minor cutting areas are shown.

The cutting area is therefore a function of δ_1 , δ_2 , ϕ , and α . The geometry for the determination of these cutting areas appears complex. If they are viewed as a section perpendicular to the direction of motion the problem is clarified; Figure 8.11 shows this.

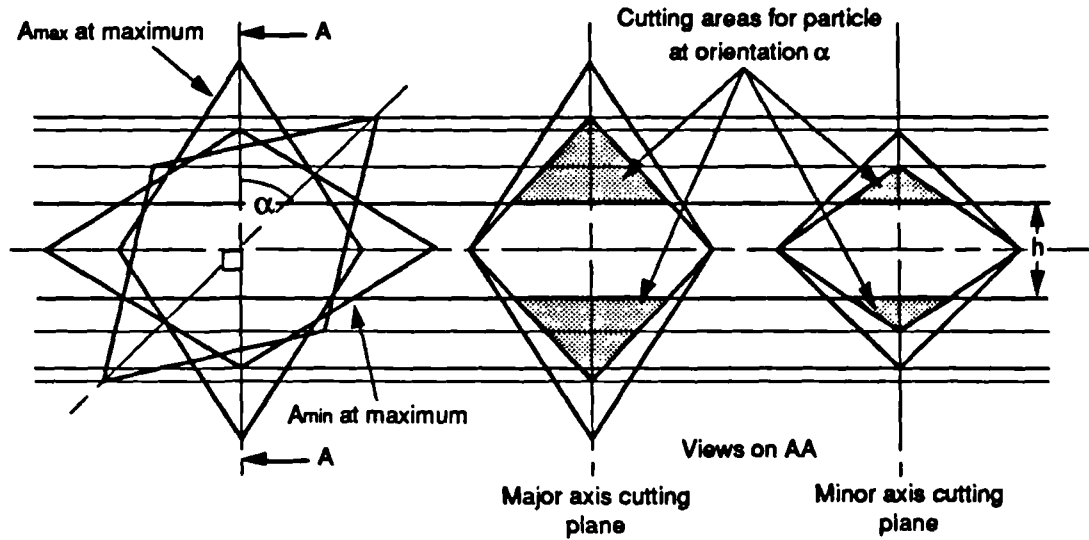


Figure 8.11. Cross sections, perpendicular to direction of motion, showing the variation in the minor and major cutting areas with orientation angle α . The figure shows three superimposed positions of the particle, $\alpha=0$, $\alpha=45$, and $\alpha=90$.

So, if A_{max} is the cutting area of the major particle axis, and A_{min} is that from the minor particle axis, trigonometry gives;

$$A_{max} = \frac{\delta_2 (\delta_1 |\cos \alpha| - h)^2}{4 \delta_1 |\cos \alpha|} \quad \text{if } \delta_1 |\cos \alpha| < h \quad \text{then } A_{max} = 0 \quad (8.26)$$

$$A_{min} = \frac{\tan \phi}{4 |\sin \alpha|} \left(\delta_2 \frac{|\sin \alpha|}{\tan \phi} - h \right)^2 \quad \text{if } \delta_2 \frac{|\sin \alpha|}{\tan \phi} < h \quad \text{then } A_{min} = 0 \quad (8.27)$$

The largest of these two displaced areas, at any given orientation angle α , will be the controlling cutting area. This value is used in the determination of the groove dimension and wear mass.

These relations are plotted in Figure 8.12; the major and minor cutting areas vary with the particle orientation; the maximum value is shown.

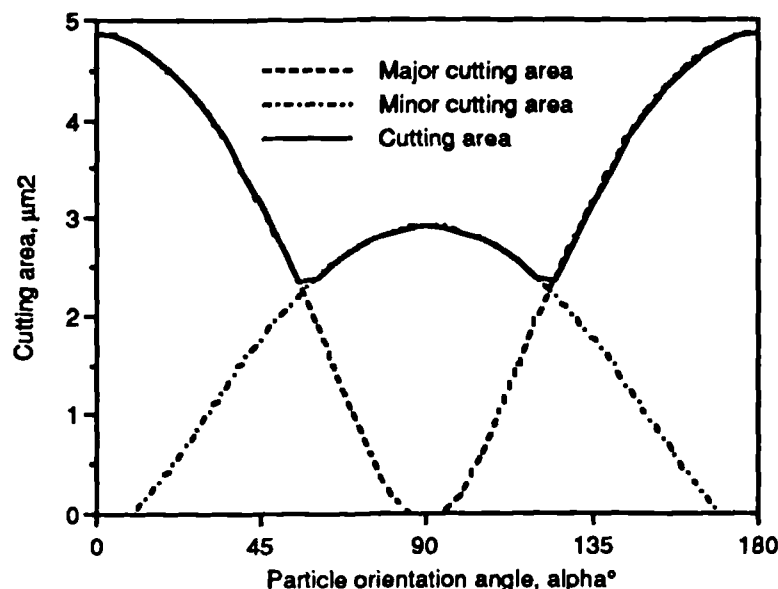


Figure 8.12. Major and minor cutting areas for a $6\mu\text{m}$ by $4\mu\text{m}$ abrasive particle orientated at angle, α° to the vertical. The resulting maximum cutting area is also plotted.

In general, the particle orientation angle, α is not known. We may use some average value to give the cutting area for the model. It is likely, however, that particles will orientate in some minimum energy position in the contact. So it is probably more reasonable to use an orientation angle of the particle which leads to a low displaced area (say $45^\circ < \alpha < 135^\circ$).

8.3.3. Number of Particles Overrolled

For an estimate of the wear volume it is essential to know the number of particles which are involved in the abrasive process. In the rolling bearing model, we assume that it is only particles that are overrolled which result in any wear (i.e. no erosive wear occurs on the free surfaces).

To determine the number of particles entrained we need a measure of the amount of lubricant overrolled. The first step, therefore, is to determine the volume swept by a single ball.

It is helpful to visualise a ball as it travels around the bearing. The area of contact it makes with the inner raceway is the locus of individual contact patches (see Figure 8.13 and 8.14).

The shaft load is supported by several balls, the maximum ball load is at the bottom ball position, $\psi=0$. The load then diminishes to zero³⁶ at $\psi=\pm\pi/2$. An additional centrifugal load on the balls is added to the outer raceway contact. Figure 8.14 shows the developed locus of contact patches for the inner raceway and outer raceway contacts. The centrifugal force on the balls leads to a non-zero contact size in the 'unloaded' regions.

³⁶ All these calculations are performed with a bearing of nominally zero diametral clearance. If there is clearance, the load falls to zero at angles less than $\pi/2$.

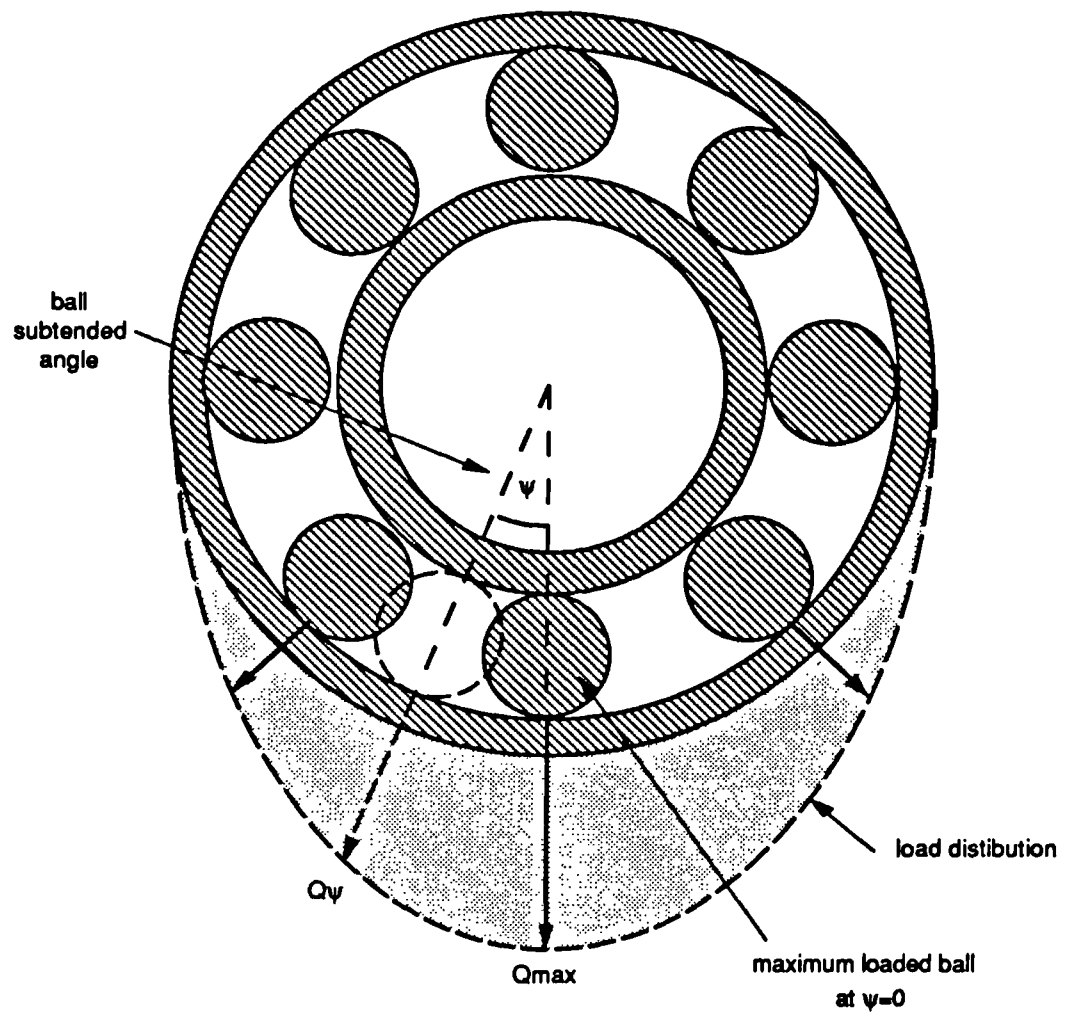


Figure 8.13. Load distribution in a zero clearance ball bearing.

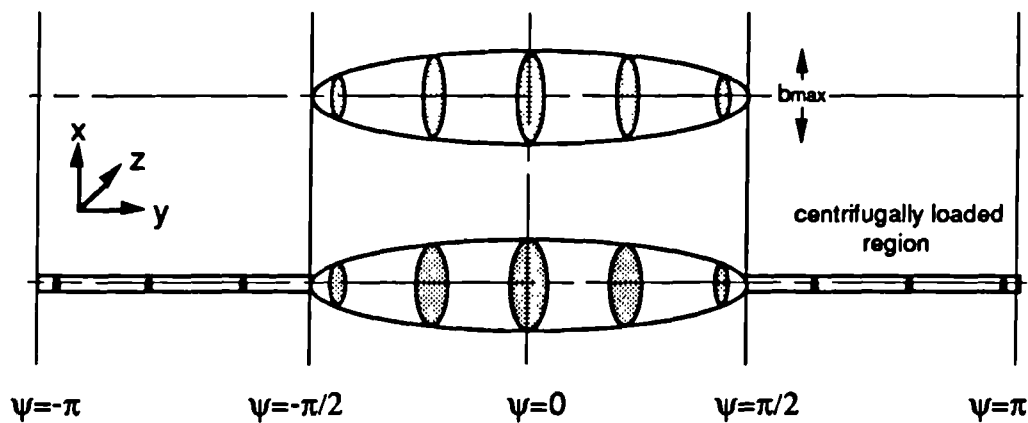


Figure 8.14. Developed locus of contact patches for the (a) inner raceway and (b) outer raceway from a zero clearance ball bearing.

The volume of lubricant overrolled is this locus area multiplied by the film thickness. The film thickness also varies with load and hence ball angle ψ .

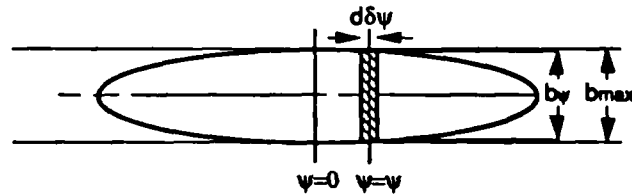


Figure 8.15. The inner raceway contact patch locus showing the variables used in the determination of the volume of swept lubricant.

First consider this locus of contact areas for the inner raceway contact. Referring to Figure 8.15, for an elemental strip of subtended angle $\partial\psi$ the volume of the strip is given by;

$$\partial V = h_{\psi} b_{\psi} d_i \partial\psi \quad (8.28)$$

where d_i is the diameter of the inner raceway contact track; b_{ψ} and h_{ψ} are, respectively, the contact width and film thickness at angle ψ . Integrating gives;

$$V = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} h_{\psi} b_{\psi} d_i \partial\psi \quad (8.29)$$

Both the contact patch and the film thickness vary with load. Hertzian analysis gives (see for example Palmgren 1945) a relation between the contact width and the load;

$$b = e_b \epsilon_E \sqrt[3]{\frac{Q}{\Sigma \rho}} \quad (8.30)$$

where Q is the load supported by a contact of width b , and the other parameters are constant for a particular bearing geometry. The dimensions of the contact patch are proportional to the load to the third root; So

$$b_{\psi} = b_{\max} \left(\frac{Q_{\psi}}{Q_{\max}} \right)^{\frac{1}{3}} \quad (8.31)$$

Stribeck's equation (see for example Harris 1984) gives a relationship between the load on the bottom ball, $Q_{\max}$ and the load at each ball position, Q_{ψ} ;

$$Q_{\psi} = Q_{\max} \left[1 - \frac{(1 - \cos \psi)}{2\epsilon} \right]^n \quad (8.32)$$

where for a zero clearance ball bearing $n=1.5$, and $\epsilon=0.5$. This gives;

$$\frac{Q_{\psi}}{Q_{\max}} = (\cos \psi)^{\frac{3}{2}} \quad (8.34)$$

Substituting this into equation (8.31) above gives;

$$b_{\Psi} = b_{\max} (\cos \Psi)^{\frac{1}{2}} \quad (8.35)$$

A similar analysis can be performed for the film thickness to determine its variation with ball angle. This time Dowson and Higginson's equation is used (see §2.1.2 for a full description);

$$H_c = 2.69 U^{0.67} G^{0.53} W^{-0.067} (1 - 0.61 e^{-0.73 k}) \quad (8.36)$$

Here, the film thickness is proportional to the load to the power -0.067. Using Stribeck's equation gives;

$$h_{\Psi} = h_{\max} (\cos \Psi)^{-0.1005} \quad (8.37)$$

Equations (8.35) and (8.37) are then substituted into the integral (8.29) to give;

$$V = d_i b_{\max} h_{\max} \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} (\cos \Psi)^{0.3995} \partial \Psi \quad (8.38)$$

This is solved numerically giving;

$$V = 0.798 \pi d_i b_{\max} h_{\max} \quad (8.39)$$

The analysis is repeated for the outer raceway locus. In this instance, for $-\pi/2 < \Psi < \pi/2$ the relation equation (8.39) holds. In the unloaded region the contact patch is of constant width, b_{cent} and given by the centrifugal force, F_{cent} ; so

$$\frac{b_{cent}}{b_{\max}} = \sqrt[3]{\frac{F_{cent}}{Q_{\max}}} \quad (8.40)$$

The centrifugal force is readily determined from the ball speed and mass (see Harris 1984). The full integral (equation 8.29) has an additional term;

$$V = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} d_o b_{\max} h_{\max} (\cos \Psi)^{0.3995} \partial \Psi + \int_{\frac{\pi}{2}}^{\frac{\pi}{2}} d_o b_{cent} h_{cent} \partial \Psi \quad (8.41)$$

This time, numerical integration gives;

$$V = 1.016 \pi d_o b_{\max} h_{\max} \quad (8.42)$$

Equations (8.39) and (8.42) represent the volume swept by a single ball as it passes around the bearing. If there are z balls then the number of ball overrollings on the inner raceway, n_{ri} , and for the outer race, n_{ro} , per bearing inner race revolution is given by (Palmgren 1945);

$$n_{ri} = 0.5 z \left(1 + \frac{2R}{d_m} \right) \quad (8.43)$$

$$n_{ro} = 0.5 z \left(1 - \frac{2R}{d_m} \right) \quad (8.44)$$

where R is the ball radius, and d_m is the contact pitch diameter. If there are n revolutions of the inner raceway in a test or bearing application, then the total swept volumes for the inner and outer raceways are;

$$V_{swepti} = 0.399 \pi d_i b_{\max} h_{\max} n z \left(1 - \frac{2R}{d_m} \right) \quad (8.45)$$

$$V_{swepto} = 0.508 \pi d_o b_{\max} h_{\max} n z \left(1 - \frac{2R}{d_m} \right) \quad (8.46)$$

These two relationships for the swept volume of lubricant for the inner and outer race contacts are then used to determine the number of overrolled particles.

From the optical ehl work it was noted that the entry of particles into the contact appeared to be as a direct ratio of the concentration of the particles in the lubricant. Assuming that this relationship holds we can arrive at a figure for the number of particles which become overrolled in a single ball pass.

This swept volume is made composed of a mixture of oil and debris so,

$$V_{swept} = V_{oil} + V_{debris} \quad (8.47)$$

If the concentration of debris in the oil is x g/l. Then;

$$V_{debris} = \frac{x}{\rho_{debris}} V_{oil} \quad (8.48)$$

and substituting for V_{oil} ;

$$V_{debris} = \frac{x}{(x + \rho_{debris})} V_{swept} \quad (8.49)$$

The volume, $V_{particle}$ of a single debris particle is found from the geometry of the model grit;

$$V_{particle} = \frac{\delta_1 \delta_2^2}{6 \tan \phi} \quad (8.50)$$

The number of particles, N in the swept volume is found by dividing the total volume of debris present by the volume of a single debris particle.

$$N = \frac{x}{(x + \rho_{debris})} V_{swept} \frac{6 \tan \phi}{\delta_1 \delta_2^2} \quad (8.51)$$

The swept volume is given by equation (8.45) for the inner raceway and (8.46) for the outer raceway; these are substituted into equation (8.51) as appropriate. The total number of debris particles overrolled during the test is then used as a multiplicative factor for the wear caused by one particle.

8.3.4. Proportion of Sliding Accommodated on Wearing Surface

One of the surprising results of the abrasive wear tests described in the previous chapter was that when using the standard ball and disk specimens, the grooves were formed on the ball whilst the disk was only pitted. This was attributed to the debris being held in the soft surface whilst it scored the harder.

In the model a parameter, g is introduced to account for this behaviour. This groove ratio, g is defined as the ratio of the length of the groove on one surface to the total sliding distance between both surfaces. So, if the length of the grooves on surfaces 1 and 2 are G_1 and G_2 respectively, then;

$$g_1 = \frac{G_1}{G_1 + G_2} \quad \text{and} \quad g_2 = \frac{G_2}{G_1 + G_2} \quad (8.52)$$

Then the parameters, g_1 and g_2 are then used to ratio the sliding distance, d_{av} (given by equation 8.25) between the pair of wearing surfaces 1 and 2 such that;

$$d_1 = g_1 d_{av} \quad \text{and} \quad d_2 = g_2 d_{av} \quad (8.53)$$

The ball on disk tests provide a convenient method for determining this parameter. For the conditions of a ball on a raceway the value of g has been estimated as 0.9 (i.e. 90% of the sliding is accommodated by groove formation on the ball).

This is in contrast to Kragelsky's (1991) assumption that the particle motion is inversely proportional to the hardness of the surfaces (as shown in equation 6.10).

8.3.5. Proportion of the Groove Removed as Wear

Multiplication of the cutting area of a particle by the sliding distance will give the amount of material displaced by the trapped particle. This, however, is not necessarily all removed from the surface of the material as a wear chip. Some of the displaced volume can be accommodated by elastic distortion of the imbedding surface; and some by plastic flow of material and the generation of shoulders to the sides of the groove.

In general, then;

$$V_{\text{disp}} = V_{\text{elastic}} + V_{\text{shoulders}} + V_{\text{chip}} \quad (8.54)$$

It is the volume of the chip formed that is of interest in the prediction of abrasive wear. It is convenient to write;

$$V_{\text{chip}} = f V_{\text{disp}} \quad (8.55)$$

where f is the proportion of an abrasive particle groove removed as a wear chip. It is this f factor which is at the heart of the abrasive wear modelling process.

The survey of literature on two-body abrasive wear has shown some important parameters which affect this material removal factor;

- (a) Material hardness, or perhaps more specifically work hardening rate. A material with the ability to work harden has a higher material removal factor, f (see Figure 6.7).
- (b) Prow angle and attack angle of the abrasive particle. High attack angles and promote chip formation (see Figure 6.8).
- (c) The critical attack angle. There is a critical attack angle below which cutting will not occur; this angle is dependent on the hardness of the surface. The harder the steel (and therefore the less it work hardens) the lower this critical angle.

Bearing steel is hard (approximately 840V) and does not work harden much; so the critical attack angle is likely to be between 30° – 40° (from Hokkirigawa 1988). The photograph included as Figure 8.6, shows the abrasive grit particles, most included angles are in the range 30° – 120° . Figure 8.16 shows the distribution of grit angles for $6\mu\text{m}$ silicon carbide powder³⁷. The data is compared with measurements of grit attack angles for abrasive paper³⁸ (Mulhearn and Samuels 1962) and for a grinding wheel surface (Doyle and Samuels 1970).

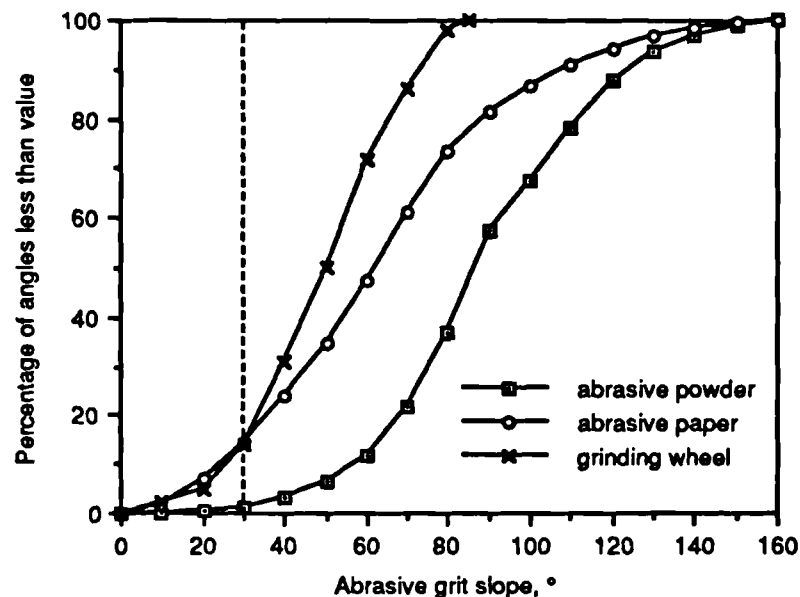


Figure 8.16. Distribution of grit angles for $6\mu\text{m}$ silicon carbide powder (plotted as % less than the angle). Compared with data for the attack angles of a silicon carbide abrasive paper and grinding wheel. The dotted line represents a typical critical attack angle.

In the third body abrasion process, the attack angle of the grit will depend on the angle at which it imbeds, or equilibrates, between the two surfaces. For these conditions f is likely to vary from 0 to 0.8 as the attack angle increases from between 30° and 40° to 90° .

These studies on two body wear give an indication of the value of the f factor. It is difficult to translate this into values meaningful for three body abrasive wear. Depending on assumptions about the orientation of a particle the value of f might vary between 0 to 0.8. It is a more satisfactory approach to use a value of f found from the test data.

³⁷ The data was obtained by enlarging the photograph and measuring the grit angles; sample size 100.

³⁸ The paper is manufactured by electrostatically causing the powder to stand on end, giving greater attack angles.

Careful measurements of the scratches generated in the ball on disk experiments give an idea of the cross sectional area of the material removed as wear (see §7.2.2.7). The f factor was calculated from this area divided by the area displaced by the particle (or the cutting area). This gave an estimate of 0.41 for silicon carbide abrasive particles. The factors for various size diamond abrasive powders were of a similar order ($f=0.28-0.51$). Unfortunately factors for further abrasive materials were not determined; the process would have involved further experimentation which time did not allow.

Another approach is to use the mass loss data from the bearing tests in conjunction with the abrasive wear model. A value of f is chosen which gives the correct predicted wear mass for one set of conditions. This value is then used to predict wear under different sets of conditions. Using the data from the bearing tests using $6\mu\text{m}$ silicon carbide abrasive gives a value of $f=0.1$.

The reasons for the discrepancy between the results from the two sets of tests, is not immediately apparent. It could be that there are other inaccuracies in the wear model (for example in the prediction of the number of overrolled particles) which lead to an underestimated mass loss.

8.3.6. Threshold for Particle Tumbling

It was noted in the ball on disk experiments with small size abrasive powder, that the particles could tumble through the contact. In §7.2.2.4 an experiment was described where particles were tumbling through the contact centre, where the film was thick, and ploughing grooves at the edges, where the film was thinner.

Williams (1992) suggests that particles of size, D to film thickness, h ratio less than a critical value will tumble through the contact. This critical value is given by;

$$\left(\frac{D}{h}\right)_c = \sec \beta \quad (8.56)$$

where β is some characteristic grit semi-angle. The results of his experiments suggest that at ratios below about 2 tumbling occurs. The paper then goes on to predict the equilibrium angles of particles trapped between two surfaces. The angles of the silicon carbide abrasive grits were measured and displayed in Figure 8.16. An average grit angle was 95° giving a critical film thickness ratio of 1.48.

For this study this critical ratio of 1.48 is used in the modelling. Time constraints restricted further experimental and theoretical research into the value of this parameter.

8.4. The Formulation of the Model

The equations derived in the preceding size sections are now combined to form a model for the abrasive wear process.

8.4.1. Determination of the Wear Mass

The final mass loss for a single contact is determined by the multiplication of the f factor, cutting area, number of particles, and average sliding distance attributable to the surface. There

is more than one contact in the ball bearing. The total mass lost is the summation of the individual mass losses for each of the contacts within the bearing;

$$M = \sum_{\psi=0}^{\psi=2\pi} M_{\psi} \quad (8.57)$$

where M is the total mass lost from the bearing and M_{ψ} is the mass lost from the ball in contact with the race at angle ψ . So for surface 1, we can determine $M_{\psi 1}$ from the multiplication of the abrasive wear factors;

$$M_{\psi 1} = f N_{\psi} d_1 A_{\psi} \rho_s \quad (8.58)$$

where N_{ψ} is the number of particles which enter the contact at angle ψ . The total mass lost is given by the integration of this;

$$M_1 = \int_0^{2\pi} f N_{\psi} d_1 A_{\psi} \rho_s \partial \psi \quad (8.59)$$

In §8.3.3 above we determined the total number of particles overrolled, N ; this is the summation of the number from each individual contact, N_{ψ} . To simplify the calculation we assume the cutting area does not vary with the ball position angle (this is a reasonable approximation since the film thickness only varies slightly and is small compared with the particle size). Hence $A_{\psi}=A$ which is constant with angle ψ , and we have;

$$M_1 = f N A \rho_s \int_0^{2\pi} d_1 \partial \psi \quad (8.60)$$

We have a relation for the sliding distance at a contact (equation 8.25) so;

$$M_1 = f N A \rho_s g_1 \frac{\pi}{20 R^2} \int_0^{2\pi} a_{\psi} b_{\psi}^2 \partial \psi \quad (8.61)$$

Equation (8.35) gives the variation of a and b with the ball angle, ψ . So;

$$M_1 = f N A \rho_s g_1 \frac{\pi a_{\max} b_{\max}}{20 R^2} \int_{\frac{\pi}{2}}^{\frac{\pi}{2}} (\cos \psi)^{\frac{3}{2}} \partial \psi \quad (8.62)$$

this is solved numerically to give;

$$M_1 = 1.748 f N A \rho_s g_1 \frac{\pi a_{\max} b_{\max}}{20 R^2} \quad (8.63)$$

This mass, M_1 is the mass lost from element 1 of the contacting pair. This calculation must be performed four times for each element in the contacting pairs of ball on inner raceway and ball on outer raceway. This gives four wear masses, the two for the ball surface are added to give, M_b (neglecting, for the moment, the ball/cage contact) and the remaining give M_{ir} for the inner race and M_{or} for the outer race. The summation of these four masses will give the total bearing mass lost .

8.4.2. Determination of the Bearing Clearance Increase

The result of this abrasive mass loss is to cause an increase in the dimensional tolerance of the bearing components. The diametral clearance in a ball bearing is given by (Harris 1984);

$$P_d = d_o - d_i - 2D \quad (8.64)$$

It is possible to estimate the change in bearing component dimensions from the mass lost.

For the ball wear we assume that the material is removed uniformly from the whole surface. If the balls maintain their trajectory in the bearing (i.e. they do not spin but track) then the wear will occur in a single band and this method will underestimate the clearance change.

If P_b is the increase in bearing clearance associated with the ball³⁹; then from the geometry of the ball;

$$P_b = \frac{4 M_b}{z \pi D^2 \rho_s} \quad (8.65)$$

Where M_b is the mass lost from all the balls and z is the number of balls in the bearing.

For the raceways we assume that the material is removed as an annular ring of width equal to the contact patch width, b . The clearance change from the outer race, P_o , and from the inner race, P_i are related to the mass lost M_o , and M_i by;

$$P_o = \frac{2 M_o}{\pi b_o d_o \rho_s} \quad (8.66)$$

$$P_i = \frac{2 M_i}{\pi b_i d_i \rho_s} \quad (8.67)$$

The diametral clearance is then the summation of these three relations (equations 8.65, 8.66, and 8.67). So;

$$P_d = \frac{2}{\pi \rho_s} \left(\frac{2 M_b}{D^2} + \frac{M_i}{b_i d_i} + \frac{M_o}{b_o d_o} \right) \quad (8.68)$$

In §7.3.2.9 predictions of the bearing clearance from masses lost are compared with the measured values. In general the results compare well but with some over prediction (see Table 7.8). This is because the annular wear of the ring is not an accurate assumption. Figure 8.17 compares this method with a more realistic wear pattern.

³⁹ This is given by four times the thickness of the layer of material removed from the ball surface.

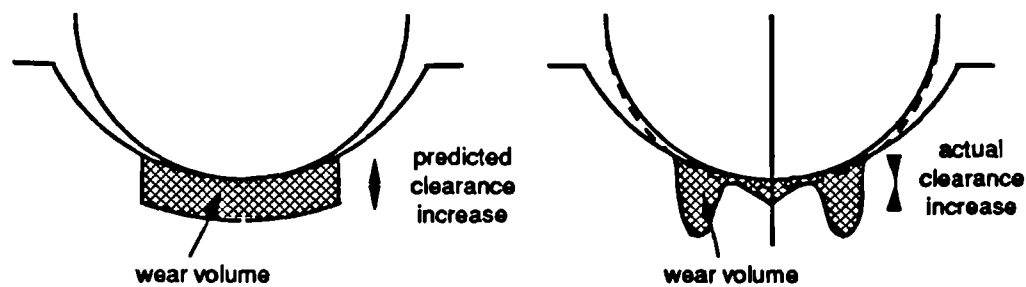


Figure 8.17. Pictorial representation of the assumed wear profile on an inner raceway. The more realistic wear profile would give a lesser increase in diametral clearance.

The ball is supported on the high spots of the worn profile and therefore shows a lower clearance increase⁴⁰.

8.4.3. The Formulation of the Model

The calculations described in the preceding sections have been incorporated into a computer program. Code was written in FORTRAN 77 and run on a Macintosh desktop computer. The program was entitled ABRASIVE.f.

The program consists of five subroutines for; input, calculation of particle entry numbers, determination of sliding distance, calculations of cutting areas, and calculation of wear masses. The film thickness and contact patch size were determined using two separate programs, FILMCALC.f and HERTZCALC.f which were linked at compilation with the main program.

Typical run times are short, since the program is merely a sequence of single equations. The advantage of using a computer program is that it can easily be adapted to loop for different values of a particular variable.

The results section demonstrates how the predicted wear mass varies with various parameters. This data was obtained by adapting the program to loop with each variable and create a data table at the final step.

8.4.4. Typical Parameter Sizes

The equations and methodology described above have been coded in FORTRAN. The program determines abrasive wear mass losses for the bearing components. A disadvantage of this 'automated' approach, is that the hand calculation of the intermediate steps are neglected; a 'feel' for the magnitude of these unfamiliar values is not obtained.

This section gives some typical values (in Tables 8.1, 8.2, 8.3, and 8.4) of the various parameters and calculated values from the analysis.

⁴⁰ In heavily loaded worn bearings it is common to find local spalling in the region of the Heathcote pure rolling bands. Figure 8.17 shows the reason for this; the load is supported on these two un-worn peaks; higher stresses and spalling results.

Parameter	Value	Units	Equation
minor contact width, a	0.201	mm	-
major contact width, b	2.85	mm	-
film thickness, h	0.6	μm	-
ball diameter, D	11.5	mm	-
raceway contact diameter, d	32	mm	-
bearing rotational speed	5000	rpm	-
mean surface speed	4.48	m/s	-
number of revolutions	2.4×10^6	-	-
test duration	8	hours	-
bearing load	4500	N	-
maximum ball load	3178	N	-

Table 8.1. Typical values of the input parameters for the abrasive wear model.

Parameter	Value	Units	Equation
volume swept in one rev, V	0.13	mm^3	(8.39)
total volume swept, V_{swept}	1.5	l	(8.45)
abrasive concentration, x	1	g/l	-
volume of debris particle, V_{debris}	16	μm^3	(8.50)
particles overrolled per rev per ball	2703	-	-
number of ball overrollings per rev	4.57	-	(8.43)
total number of particles overrolled, N	2.9×10^{10}	-	(8.51)

Table 8.2. Typical values of the debris particle number parameters for the abrasive wear model.

Parameter	Value	Units	Equation
maximum creep ratio, ζ_{max}	+1.5	%	(8.17)
minimum creep ratio, ζ_{min}	-10.0	%	(8.17)
maximum sliding distance, d_{max}	15	μm	(8.23)
minimum sliding distance, d_{min}	0	μm	(8.23)
average sliding distance, d_{av}	7.75	μm	(8.25)

Table 8.3. Typical values of the sliding distance parameters for the abrasive wear model.

Parameter	Value	Units	Equation
cutting area for $\alpha=0^\circ$, A	4.86	μm^2	(8.26)
cutting area for $\alpha=90^\circ$, A	2.89	μm^2	(8.27)
average cutting area, A^{41}	3.51	μm^2	-

Table 8.4. Typical values of the cutting area parameters for the abrasive wear model (Figure 8.11 shows the variation of cutting area with orientation angle).

⁴¹ Determined for particles orientated at a range of angles from 0-180° (evenly distributed).

8.5. Results

Using equation (8.63) the wear mass of the whole bearing can be calculated. In this section the effects of the various parameters are investigated and compared to the results from the bearing tests (described in Chapter 7).

The value of the f factor has been determined from the bearing test data using 1g/l of 6 μ m silicon carbide abrasive. So in other words, the model has been fitted to this particular operating condition; the data at other sets of conditions are then compared with the predictions of the model.

8.5.1. Abrasive Wear Variation with Abrasive Particle Orientation

The abrasive particles are generally not symmetrical. Depending on the orientation of the abrasive particle a smaller cutting area may be presented to the surfaces (see Figures 8.10 and 8.11). It is uncertain whether they orientate to a streamline position; in the entry region or contact. Figure 8.18 shows the predicted wear mass against the orientation angle of the particles.

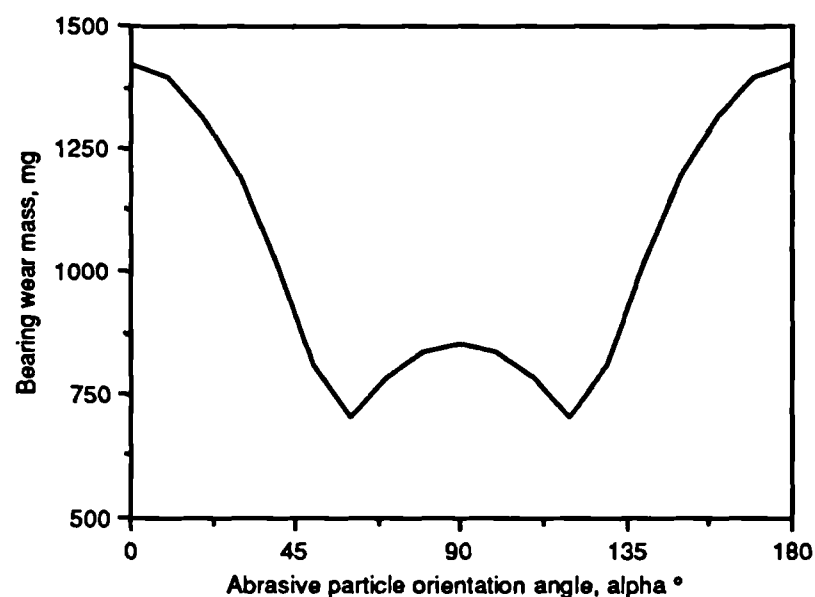


Figure 8.18. The variation of predicted bearing mass loss with abrasive particle orientation angle (for 1g/l of 6 μ m silicon carbide abrasive).

A factor of 2 exists between the mass loss when the particle enters at upright position compared to that when the particle orientates to a more streamline position. For particles of a similar size to the film thickness this ratio can be considerably higher.

The choice of an orientation angle is thus important in the determination of abrasive mass loss. It might be expected that the particles will orientate at a 'minimum energy position' and so result in a lower wear mass than predicted using an average value.

In the analysis in the following sections the particles were assumed to orientate in the contact. An orientation angle, α was chosen to give a minimum cutting area to determine the wear mass (i.e. in the graph above an angle of 60° or 120° would have been used).

8.5.2. Abrasive Wear with Debris Concentration

The model predicts a linear increase in bearing mass loss with increasing concentration. The wear is directly proportional to the number of particles overrolled (each particle is assumed to act additively to the final wear mass) which is in turn proportional to the debris bulk concentration.

Figure 8.19 shows the model predictions alongside the test data. On the basis of the experimental data scatter, a 50% error bar is given to the wear masses.

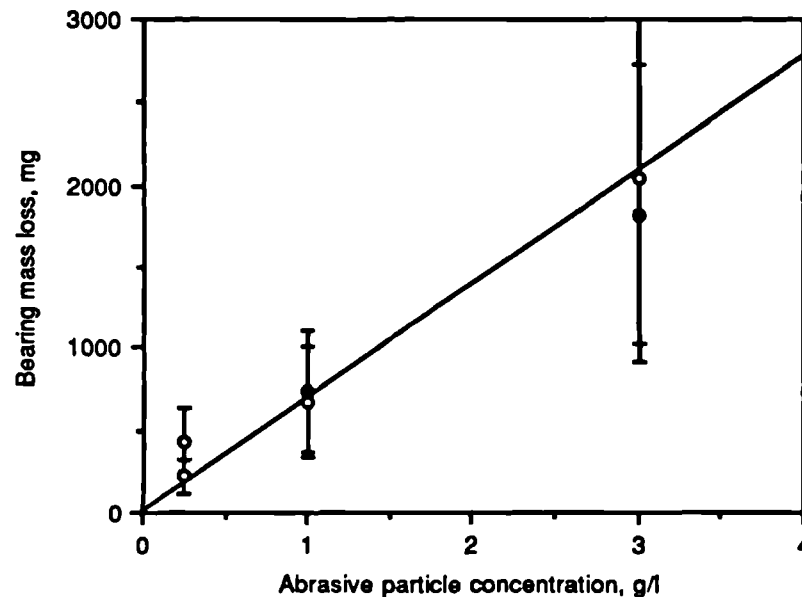


Figure 8.19. The variation of bearing mass abrasive loss with concentration of abrasive particles. The results of the model (the model has been fitted to the data points at 1 g/l concentration) are compared with bearing wear test data.

The test data does indeed show a linear variation with debris concentration. The test data is well predicted by the model.

8.5.3. Abrasive Wear Variation with Load

Increasing the load on a ball bearing causes the size of the contact patches to increase; as discussed before, the particles are fully imbedded in the two surfaces and so a change in load does not effect the cutting area of an abrasive grit (unlike a two-body wear process).

The load therefore acts to increase the number of particles entrained into the contact; and increases the distance each particle slides.

Figure 8.20 shows the model predictions of abrasive wear with increasing load. Reasonable correlation is seen for the other load data point. There is not enough test data to draw any real conclusions about the validity of the model.

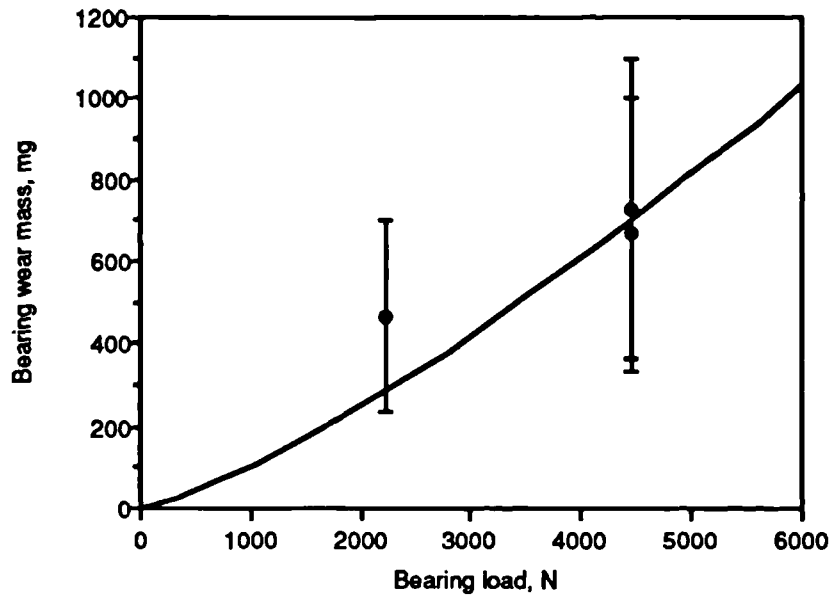


Figure 8.20. The variation of bearing mass abrasive loss with bearing load. The results of the model (the model has been fitted to the data points at 4500N load) are compared with bearing wear test data.

Equation (8.63) shows that the wear mass is proportional to the number of particles and the sliding distance. The other variables vary only slightly with load (i.e. the film thickness and cutting areas), So;

$$M \propto N d_{av} \quad (8.69)$$

and in turn using equations (8.25) and (8.51) gives;

$$M \propto ab^3 \quad (8.70)$$

Hertz analysis states that the contact widths are proportional to the load to the cube root, so;

$$M \propto L^{\frac{4}{3}} \quad (8.71)$$

The model predicts a wear mass proportional to the bearing load to the power of 4/3 (re plotting Figure 8.18 as a log/log plot also shows this). Further experimentation is required to verify this prediction.

8.5.4. Abrasive Wear Variation with Abrasive Particle Size

As the size of the debris particle decreases so does the cutting area. However the number of particles, for a given debris concentration (in g/l), increases dramatically. Figure 8.21 shows the variation of cutting area and number of particles entering the contact.

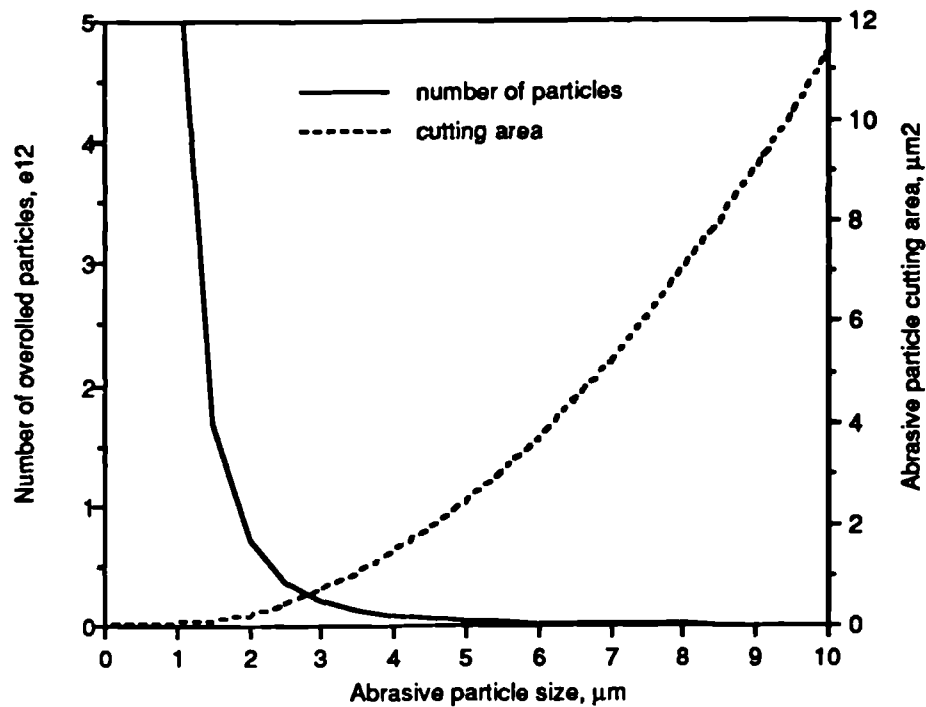


Figure 8.21. The variation of particle cutting area and number of particles entering with abrasive particle size, as predicted by equations (8.26 & 8.27) and (8.51).

The number of particles is inversely proportional to the cube of the debris size, whilst the cutting area is proportional to the square. The abrasive mass loss is then a compromise of these two characteristics. As the particle size reduces the number of particles increases massively but their cutting area drops. Figure 8.22 shows the predicted variation in bearing wear with particle size.

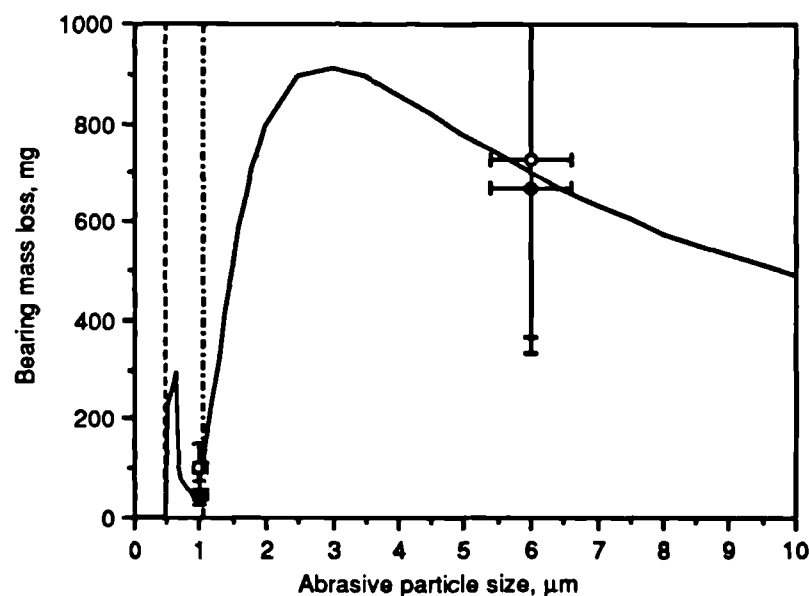


Figure 8.22. The variation of bearing mass loss with abrasive particle size. A constant concentration of debris is maintained. The results of the model (the model has been fitted to the data points for a particle size of $6\mu\text{m}$) are compared with bearing wear test data.

Two thresholds exist (shown as dotted lines); firstly the transition from ploughing to tumbling at $\delta_1=1.1$; and secondly when the particle size is less than the film thickness, at $\delta_1=0.6$.

Prediction of wear at these small particle sizes is difficult, the slopes are steep and very dependent on the choice of tumbling transition. The data points are from tests using Arizona dust (which is thought to have a fragment size of about 0.5 to 1 μm) and 6 μm silicon carbide. Further experimental data is required for a full investigation.

8.6. Limitations of the Model

The model presented here is a first attempt at the modelling of bearing wear by lubricant debris. It is only realistic under certain conditions; the limitations are given in this section.

8.6.1. The Abrasive Debris Particles

The model considers the debris as a rigid particle. The deformation of the particle by either fracture or ductile flattening is not considered. The model was designed for, and is best suited to, the modelling of the fragments of fractured brittle debris materials.

A low toughness brittle material, like glass or sand, fractures readily in the inlet region and wear will be caused by the resulting fragments. Small size particles, like 6 μm silicon carbide will not fracture in the contact and will cause wear in their original shape. The wear model is designed for this type of debris. To model the behaviour of ductile materials, which flatten out in the contact, requires further work. The cutting area is dependent on the deformed shape of the particle. Test work is required to determine the nature of the grooving process by different debris materials.

8.6.2. The f Factor

At the heart of the wear model is the proportion of the particle displaced area which becomes removed as a wear chip, the f factor. This was determined from scratch experiments and compared with values required to make the model fit the bearing test data. Some discrepancy exists (with $f=0.41$ from scratch tests and $f=0.1$ from bearing tests). This may be because some other features of the model do not correctly predict the behaviour in a rolling bearing (for example the debris entry or sliding processes). Or, the scratch tests may not correctly determine the value of the factor.

Equation (7.4) defines f as the area of the scratch minus the area of the shoulders divided by the area displaced by the grit particle, $(A_3-A_2)/A_{disp}$. No account is made for the proportion of the displaced area which plastically flows into the side walls of the scratch. This would reduce the value of f . Some plastic flow or finite element indentation calculations are required to solve this problem.

It is the authors view that this sort of three body scratch test is a good method for determining how a debris particle removes material from a rolling element. In these tests only a few debris materials were tested, more information is required. The f factor determined from the silicon carbide or diamond abrasive tests may not be applicable for other particles. In addition, f may be dependent on other variables; rolling element material, humidity, particle sharpness etc.

It is envisaged that for each set of conditions of contaminated lubricant and bearing operation, a ball on disk test would be run. Investigation of the scratched surfaces would determine the

abrasivity (i.e. f) of the lubricant debris under those conditions. This would then be used in the bearing wear model.

8.6.3. The Role of the Cage

The model presented takes no account of abrasive wear action between the cage and the balls. The bearing tests suggested that the majority of mass loss, for plastic cage bearings, was caused by the interaction between the balls and the raceways. However, different cage materials may not have the same result.

The modelling of the wear caused by the cage is difficult, largely because the kinematics are complex. To model the wear process a prediction of the following parameters would be required;

- (a) The ball/cage contact area. The maximum contact area could be determined from the wear patch on the cage (see Figure 7.25).
- (b) The sliding distance. When the ball and cage are in contact the surfaces are 100% sliding. However, this sliding distance will change with angular position of the ball. For one half of the ball passage it is driven by the cage, for the other half it drives. A relation for this variation of sliding distance with angular position is required.
- (c) The relative imbedment of the particles. The softer cage material will imbed debris particles to some extent; this must be quantified to give a grit cutting area.
- (d) The breakdown of particles in a cage/ball contact. The model presented here considers wear by final fragments of debris broken down in a steel on steel contact. Wear in a cage/ball contact may not be controlled by the same size fragments.

It is beyond the scope of the work to study this problem further.

8.6.4. Applicability To Other Tribological Configurations

The wear model has been written in a general form. The basis of the model should stand for any closed three body wear process. Particles are trapped between surfaces and become fully imbedded. A knowledge of the number of particles to enter and the sliding and contact conditions are required. The wear of gear teeth, for example, by lubricant debris could be modelled using this approach.

8.7. Conclusions

A model to predict abrasive wear in rolling bearings by lubricant debris has been devised. The model considers the wear at the ball and raceway contacts; the effect of the cage is not considered.

The wear mass is determined by the multiplication of the following parameters;

- (a) number of particles which enter a contact,
- (b) the abrasive particle cutting areas,

- (c) the distance each particle slides,
- (d) the mechanism by which the particles scratch the surfaces,
- (e) the proportion of material removed by each particle.

The first three of these parameters are determined from the kinematics of the bearing and assumptions about the particle geometry. The last two are based on experimental evidence from the ball on disk tests and bearing abrasive wear tests described in Chapter 7.

The data from one set of bearing tests has been used to determine constants in the model. The model has then been used to predict the results at other test conditions. A reasonable correlation is seen; the behaviour of wear with debris concentration and load is well predicted. The relationship between wear and particle size is still somewhat uncertain.

CHAPTER 9

CONCLUSIONS

9.1. The Contamination Performance Relationship

This work has been aimed at improving the understanding of the relationship between lubricant contamination and the failure of rolling element bearings. The relation requires knowledge of the types of contaminating particles which may be present in practical lubricants; how the particles behave in the contact and entry zone, a quantification of the nature of the surface damage, and how this surface damage leads to rolling contact fatigue or abrasive mass loss.

The project has been approached by a combination of experimental and theoretical study. Optical ehl was used to visualise debris particles in and around a rolling lubricated contact. The surface damage on a pair of rollers in the mangle test rig was related to the debris particles present and their mechanism of deformation in the entry region.

The failure of debris damaged surfaces by rolling contact fatigue has been investigated using a theoretical approach. The analysis was performed using a finite element indentation model in conjunction with a discretised fatigue life model.

Three approaches to the study of abrasive wear failure have been carried out. Firstly, some fundamental tests, using a simplified bearing type contact, were performed to demonstrate how particles behave under conditions of relative sliding and remove material from surfaces. Secondly, the wear of deep groove ball bearings by abrasives in the lubricant has been investigated. On the basis of these experimental studies a model of ball bearing abrasive wear has been developed.

9.2. A Summary of the Findings

This section contains, in note form, the main findings from each of the individual studies.

9.2.1. Debris Behaviour in a Concentrated Contact

The optical elastohydrodynamic apparatus proved a useful tool for the study of lubricant debris particles in and around a model contact. The following general points have been concluded from the test work;

- (a) Ductile materials flatten in the entry zone whilst brittle material fracture; the resulting platelet or fracture fragments enter the contact.
- (b) The particles did not upset the lubricant film; loads on the particles caused them to be fully imbedded between the surfaces.
- (c) The number of particles entering the contact appeared to be approximately in proportion to their concentration in the bulk lubricant.

9.2.2. Debris Initiated Surface Damage

The 'mangle rig', based on pair of contacting rollers, was used to study the relationship between lubricant debris and surface damage. The following conclusions are made;

- (a) The scale of damage caused by ductile and brittle particles is dependent on the particle hardness and fracture toughness respectively.
- (b) Ductile debris damage can be modelled on a large scale by squashing pellets between anvils.
- (c) Predictions on the size of fracture fragments to survive a rolling contact can be made on the basis of these experiments.

9.2.3. The Stresses and Deformations Associated with Indentation

The finite element method was used to model the elastic plastic indentation of a half space. Elastic, plastic, and residual stresses and deformations were investigated for several models. The following general points are noted;

- (a) Elastic and elastic-plastic solutions agreed well with data from the work of other workers.
- (b) Residual stresses and indentation shapes after unloading were generated; the accuracy of the results showed some sensitivity to the loading steps.
- (c) A shallow layer of radial tension was left within the contact region after unloading.
- (d) The depths of residual indentations, even when strain hardening was modelled, were about 15% smaller than those created experimentally

9.2.4. Debris Initiated Fatigue

The residual stresses and deformations predicted from the finite element indentation model were used in conjunction with a rolling contact fatigue life model. The following points were concluded from the study;

- (a) The shape of the indentation played a major role in determining fatigue life. Deep steep sloped dents overrolled in small contacts caused the greatest reduction in life.
- (b) The residual stresses following indentation had a minor effect on the calculated life reduction. In some cases they proved beneficial.
- (c) Fatigue cracks were predicted to initiate from the trailing edge of a dent.
- (d) Threshold dent sizes and slopes exist below which fatigue was predicted not to occur.

9.2.5. Abrasive Wear of Bearings by Lubricant Debris

A ball on disk contact geometry was used to understand some of the fundamentals of the abrasive wear process and how debris particles behave in a rolling and sliding contact. A bearing test machine was used to provide some data on the wear of deep groove ball bearings. From this information a model for bearing wear, under certain conditions, was developed. The following general points are noted;

- (a) Abrasive particles were found to imbed in the softer disk surface and scratch the harder ball. The ball surfaces were covered with abrasive grooves, equal in length to the sliding distance, whilst the disk was only pitted.
- (b) Changing the hardness ratio of the elements altered the location of the grooves.
- (c) Particles were found to tumble, rather than groove, in the contact when the ratio of particle size to film thickness was less than about 2.
- (d) The distribution of wear in a ball bearing was found to concur with these mechanisms.
- (e) Bearing wear rates were found to be proportional with time and concentration.
- (f) The plastic cages used in these tests were thought to contribute little to the wear of the bearing.
- (g) The abrasive model includes parameters for; the number of particles entrained, the cutting area of each particle, the sliding distance, the proportion of sliding accommodated at the surface, a threshold for particle tumbling, and a factor to determine how much material is removed from the ploughed groove.
- (h) The theoretical model over predicts wear. Using empirical data from one set of bearing tests to provide the constants gives good predictions for other data points.

9.3. Avoiding Debris Initiated Failure

It is interesting to speculate, given the amount of effort devoted to study the problem, on ways to avoid this debris initiated fatigue or wear failure.

9.3.1. Filtration

The obvious solution is to ensure debris is not present in the lubricant in the first place. Either by ensuring a system with no possibility of debris ingestion or generation, or by adequate

filtration. Unfortunately the filtration process implies added cost, and pressure drops to be overcome. The work on relating debris to fatigue life can be of help here; understanding what is the minimum particle size to initiate fatigue can be used to design a filter rating for use in the system.

However in the abrasive wear process, we have seen how small particles can cause extreme wear. Filters to operate to this level of cleanliness are expensive and liable to rapid blocking. Here, it is important to understand whether the contaminating materials pose a problem, and if they will lead to unacceptable wear rates.

9.3.2. Bearing Design

During normal operation of a rolling bearing wear debris is not generated, and only external sources of debris are of concern. So sealing a bearing for life may provide some protection against debris damage.

Given the ease with which debris particles become entrained into a rolling contact, it is difficult to visualise some bearing element or cage design which may reduce the debris entrainment potential. Cages with sweeping arm design are manufactured; these are aimed at removing very large solid particles from the ball path, in conditions where the contamination is so excessive that seizure may occur.

9.3.3. Built-in Redundancy

Knowing the life of a rolling bearing under given conditions of lubricant contamination can allow for the built in redundancy of the component; this planned maintenance operation is generally cheaper than having plant under emergency shut down. The process will require lubricant condition monitoring for both debris material and size information.

9.3.4. Materials

The abrasive wear experiments performed here and the debris indent experiments performed by Hamer and Hutchinson (1992) have shown the importance of the rolling element hardness ratio. It may be possible, using this information, to choose a rolling element hardness ratio to locate the indentations or wear scratches on a particular element and thereby achieve optimum life.

9.3.5. Abrasive Wear vs Fatigue

It may be possible to use the abrasive wear process to reduce the likelihood of debris indent fatigue. Fatigue failure generally occurs long before wear failure. Using fine ceramic particles to abrade out the large indentations caused by ductile debris overrolling may shift the failure mechanism from short fatigue lives to a longer wear life.

9.4. Further Work

The following list contains some ideas which could be carried out, using existing equipment, to further the understanding of the process; and aid in the development of the model.

- (a) Ball on disk tests to determine the wear process, and f factors for other types of debris material; including different size particles and ductile materials.
- (b) Bearing test data for a range of abrasive particle sizes, to fully explore the relationships between number of particles, cutting area, and tumbling threshold.
- (c) Bearing tests or ball on disk tests using thinner lubricant films; to model starvation effects and how this might control the motion of a particle.
- (d) Bearing tests with different cage designs and material. The existing tests were undertaken using plastic cages which did not appear to have a major effect on the mass loss.
- (e) A deeper investigation into the number of particles entering a rolling bearing type contact is required. A study based on counting debris indents for a given test time and debris concentration would be simple, but tedious, to perform.
- (f) The mechanism of debris imbedding in one surface and scratching the other is of interest was not fully investigated here. Ball on disk, and bearing tests, could be performed using different hardness ratio elements, correctly tempered and machined, for a more conclusive study.

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Appendix 1

Debris Materials, Lubricants, & Test Specimens

A1.1. Description of Debris Materials

Test contaminant materials were chosen to be a realistic reflection on the type of debris particles which might be found in a lubricant. The particles chosen were broadly divided into three categories, ductile, brittle ceramic, and tough ceramic.

Tables A1.1 and A1.2 lists the powder materials used, along with some materials properties. Where powder data is unavailable, material properties of the bulk solid have been given. It should be noted however that in some cases the properties of the fine particulate does not correspond to those of the bulk. This is particularly true for ceramic particles where the strength is dependent on particle size. The data should be considered as approximate only.

Material	Particle Size	Particle Shape	Hardness, GPa	Young's Modulus, GPa	Poisson's ratio
PTFE	4-9 μ m	Spherical	-	0.3-0.8	-
Copper	0-60 μ m	Spherical	0.5	130	0.35
Aluminium	0-100 μ m	Irregular	0.2	71	0.34
Low Carbon Steel	0-100 μ m	Irregular	1.3-2.4	207	0.29
Cast Iron	0-75 μ m	Irregular	0.3	110	0.27

Table A1.1. Materials properties of ductile debris powders used in experiments

Material	Particle Size	Particle Shape	Hardness, GPa	Toughness, MPa $\sqrt{m}$	Modulus, GPa
Arizona Dust	0-100 μ m	Irregular	10	1.5	-
Glass Ballotini	0-60 μ m	Spherical	5	0.7-0.8	71-80
Silicon Carbide	100 μ m	Cubic	20-25	4-5	410
Boron Carbide	0-60 μ m	Cubic	25-35	1.7	440-470
Alumina	0-100 μ m	Irregular	18-20	3-4	330-400

Table A1.2. Materials properties of brittle debris powders used in experiments

The particles when purchased were generally in powder form with a broad size distribution (typically 0 to 100 μ m diameter). A stack of wire mesh sieves were used to separate the batches into close tolerance bands. The sieves were graded to sizes 23 μ m, 45 μ m, 53 μ m, 58 μ m, 63 μ m, and 75 μ m. With prolonged sieving it was possible to obtain reasonable size tolerance batches.

In addition some finely graded diamond powder was obtained for abrasive wear studies. The material was synthetically manufactured diamond (deBeers grade MDA). The particles were tough and of uniform blocky size and shape. Size bands of 0-0.5 μ m, 0.5-1 μ m, 1-2 μ m, 2-4 μ m, and 3-6 μ m were obtained.

A1.2. Description of Test Lubricant

The ball on disk tests, bearing tests, optical ehl, and mangle rig experiments were performed using Shell Turbo T68 as the lubricant. This is a standard steam turbine lubricant, used at SKF ERC as a test oil. The lubricant contains some anti-oxidation additives and a VI improver. The simple, low additive, formulation makes the lubricant a good choice for studying the effects of debris particles without interference from the lubricant properties. Table A1.3 shows some of the lubricant properties.

Density at 15°C	0.876kg/m ³
Viscosity at 40°C	68 cSt
Viscosity at 100°C	8.8 cSt
Viscosity index	102

Table A1.3. Properties of Shell lubricant T68 (from Shell data sheet LPDS/86/588)

A1.3. Description of Test Specimens

The test specimens used during the various experiments were from standard rolling element bearing components. They therefore had realistic materials properties, surface finish, geometries, and were easy to obtain.

- The Mangle Rig. These specimens were cylindrical rollers, from a roller bearing. The diameter of the rollers was 15mm and their length 20mm. They were not perfectly cylindrical but had a crowning of radius 5.5m.
- The Ball on Disk Rig. The specimens were standard 1" balls from a deep groove ball bearing. The disks were the inner rings from a cylindrical roller bearing (SKF part number NU 315 ECJ) and were 95mm in diameter.
- The Bearing Test Rig. The specimens were deep groove ball bearing (SKF 6305) with polyamide 6,6 or pressed steel cages.

All specimens were of through hardened carbon chromium steel with a hardness in the range 7 to 8.5 GPa. The components are all ground with surface finishes of about $R_a=0.01$ for the balls and rollers and $R_a=0.1$ for the large races.

A1.4. Test Specimen Heat Treatment

For some of the tests it was necessary to soften the specimens. This was carried out by tempering them in a furnace for two hours and allowing to cool in air. Surface oxidation was kept to a minimum by coating the specimens in Bergatek, a special heat treating paint. The graph included as Figure A1.1 shows the furnace temperatures and the resulting hardness.

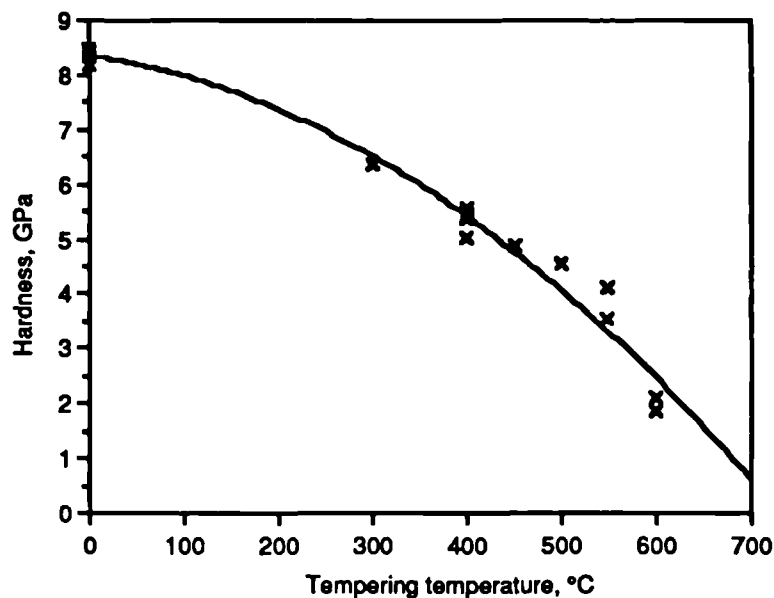


Figure A1.1. Graph of tempering temperature against resulting specimen hardness. Specimens were heated for two hours and allowed to cool in air.

A1.5. Contact Conditions in the Experimental Apparatus

The elastic contact solution of Hertz (1882) has been used to determine the contact conditions in the various test apparatus. A short program was written in FORTRAN to determine the contact dimensions. Table A1.4 contains the data.

Apparatus	Contact type	Dimensions, mm		Peak pressure, GPa
		a	b	
Optical ehl	circular	0.201	0.201	0.6
Mangle rig	elliptical	0.079	4.500	1.6
Disk machine	elliptical	0.288	0.337	2
Bearing rig ⁴²	elliptical	0.201	2.850	2.64

Table A1.4. Dimension of contacts in the test apparatus.

⁴² For the ball on inner ring contact at the position of maximum load.

Appendix 2

Determination Of Strain Hardening Properties

The measurement of the properties of bearing steel far beyond yield is difficult. Special specimens would have to be machined, heat treated, and tested in some form of torsion test.

Tabor (1951) gives a simple method for determining strain hardening properties using the Brinell hardness test. The Brinell hardness test differs from the equivalent Vickers test in that a sphere is used for indentation. So, as the depth of penetration increases the level of plastic flow increases and so the material work hardens (the Vickers indenter is a self-similar pyramid so the level of work hardening does not change with indent depth).

Tabor performed micro Vickers indentations in the region of a Brinell indentation to determine the local yield stress. He found that the strain associated with the indent could be empirically expressed as $20d/D$ (where d is the indent diameter and D is the Brinell ball diameter). And that the mean pressure was approximately 2.8 times the yield stress. So plotting these values gives a stress-strain curve. Tabor verified the coincidence of this stress-strain curve with that generated from a tensile test; agreement was exact.

This method was adopted to determine the stress-strain curve for bearing steel. A 10mm tungsten carbide ball was loaded onto the surface of a bearing sample, indentation geometry and loads were noted. The data points are plotted and a strain hardening line is fitted. This is then intersected with the elastic stress strain curve.

Figure A2.1 shows a plot of the indentation experiment results, and the predicted stress-strain curve.

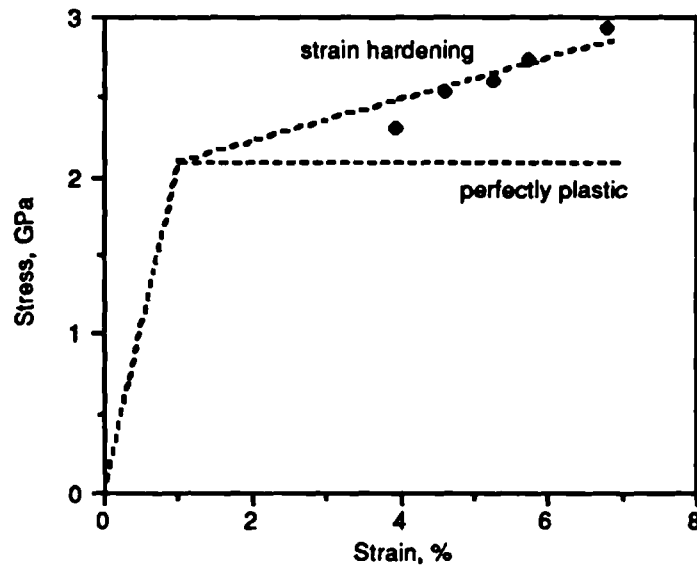


Figure A2.1. Stress-strain curve for bearing steel derived from Brinell indentation experiments (after the method of Tabor 1951).

It can be seen that the yield stress increases beyond first yield (a perfectly plastic material would have shown no increase in yield stress past first yield). The bearing steel therefore work hardens to a surprisingly large degree. It is this strain hardening data which was used in the finite element analysis description of the bearing steel.